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Calibration of Eccentric and Segmental Orifices in 4 and 6-In. Pipe Lines

By S. R. BEITLER[1] and D. J. MASSON,[2] COLUMBUS, OHIO

This is a preliminary report of a Subcommittee on Eccentric and Segmental Orifices of the Special Research Committee on Fluid Meters on the Calibration of Eccentric and Segmental Orifices. It describes the method used in calibrating orifices in 4 and 6-in. pipe lines and gives the results of the calibration which are available at this time. This program is only in its early stage and as a result, about 50 per cent of the work has been done.

THE standard primary devices used for flow measurement in pipe lines are Venturi tubes, flow nozzles, and sharp-edged concentric orifices. These devices have all been described adequately, and a great deal of research work has been carried out to determine their flow characteristics and coefficients of discharge. The results of these researches have been published by the Society and other interested groups, and the analysis of the results of the researches have been made into suggested standards for computing flow measurement, using the devices mentioned.

In many cases where the fluids to be measured are nonhomogeneous, the standard measuring devices have certain drawbacks which make other metering elements more desirable. Two of these elements are sharp-edged orifices. One is the form in which a round orifice is made in a plate in such a way that one edge of the orifice is practically tangent to the wall of the pipe line, so that there is no obstruction to flow on one side of the pipe line. This can be used then for a vapor or gas carrying some liquid, or a liquid carrying some solids, if the edge of the orifice is flush with the bottom of the pipe line. The extraneous material will then pass through the orifice without building up in front of the orifice plate. If the extraneous material is a gas in a liquid, the tangential side is put at the top of the pipe, and the gas will pass through without altering materially the flow condition.

A second device is an orifice plate with an orifice in the shape of a segment of the pipe into which it is to be inserted. This type is ordinarily used only when the impurities in the flowing stream are heavier than the main material being measured, so that the opening is at the bottom of the pipe. This orifice is made with a segment circle diameter—approximately 97 or 98 per cent of the diameter of the pipe—and the area varied by altering the height of the segmental opening.

It has been thought probable that these devices can be designed and manufactured so that their coefficients of discharge can be predicted and that they will be almost as satisfactory for accurate flow measurement as are the concentric orifices. With this idea in mind, the Fluid Meters Committee has set up a calibration program to determine the values of the coefficients of discharge for these primary elements over a wide range of pipe sizes, area ratios, and rates of flow, so as to be able to publish tables of coefficients for these primary elements similar to those available for nozzles and concentric orifices. The program was started early in 1948, and the work is almost completed on orifices in a 4-in. pipe line.

In addition to the foregoing, the Bailey Meter Company, in 1945 and 1946, supported a research program covering the same sort of investigation on similar orifices in 6-in. pipes, and the results of this investigation have been made available to the committee and are presented as a part of this paper.

It should be emphasized at this point that the results presented in this paper are not the final results, and that they are presented at this time in order to make the preliminary results of this investigation available to meter users as rapidly as they are secured. It is probable that when the results of calibration of similar orifices in larger pipe lines are available, the curves presented for 4 and 6-in. lines will be modified in order that a consistent set of curves covering all pipe sizes can be plotted. Experience in analyzing the results on concentric orifices and flow nozzles has shown that within the limits of ordinary measurement accuracy, it is possible to determine the variation of coefficients of discharge with pipe size, area ratio, and Reynolds number, so that a series of curves or an empirical formula can be set up which will represent the results of the researches. It is planned that this will be done as soon as the data taken cover a wide enough range to make it possible. No attempt has been made to correlate the effect of pipe sizes in the curves given in this paper, since only two pipe sizes are available, and they do not cover a very wide range of pipe diameters.

At the time of presentation of this paper, the 4-in. work was nearing completion, and it was planned to make a start on calibrating the orifices in a 10-in. pipe line. In order to complete the program, it is also planned to do work with a 14-in. pipe line. It is thought that if the range of from 4-in. to 14-in. sizes is covered completely, it will be possible to extrapolate the data for larger-sized lines, and at present it is not believed that these orifices will be very satisfactory for lines under 4 in. diam.

Arrangements have been made with the Research Foundation and the Engineering Experiment Station at The Ohio State University to carry out this research at a cost to the committee for the labor only, the university furnishing all space and power requirements, so that a considerable saving has been effected in this operation.

Method of Testing

The piping and the manometers which had been constructed at the university for calibration of concentric orifices and nozzles were already available and were used for this test work. The orifice plates which have been used to date are made of $^1/_{16}$-in. monel metal and in all cases the orifices are bored with a sharp edge upstream and the downstream edge beveled to $^1/_{32}$ in. edge thickness. Pins in the upstream face of the orifice, about $^1/_{16}$ in. diam., are riveted to the orifice in such a way that they locate the orifice exactly in the upstream pipe section. The orifice is mounted between flanges which are doweled so that the inlet and outlet pipes are lined up as accurately as possible.

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Contributed by The Special Research Committee on Fluid Meters and presented at the Annual Meeting, New York, N. Y., November 28–December 3, 1948, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society. Paper No. 48—A-49.

This means that when the orifice is clamped between flanges, it is then correctly located with respect to both upstream and downstream pipe sections.

The testing sections are made of standard pipe which is cleaned at regular intervals to keep the pipe surface as nearly constant as possible. These pipes were chosen from standard 4 and 6-in. pipe sections which were commercially round and smooth. The pipe lines were erected in such a way that there were at least 40 diameters of straight pipe ahead of the meter flange, and straightening vanes were mounted ahead of the 40-diam meter run so as to provide smooth straight flow to the orifices. It is probable that the velocity profile ahead of the orifice is very close to that which is considered normal for turbulent pipe flow. Pressure taps were made at 1 pipe diam and 1 in. upstream from the orifice plate, and at tenths of diameters downstream from the orifice plate, with a tap 1 in. downstream and one approximately 8 pipe diam downstream—the latter to measure the head loss in the orifice. An additional tap was placed well upstream in the 4-in. line in an attempt to measure the pipe friction factor, *f*, to see if it is possible to correlate the coefficient of discharge with this friction factor.

The manometers used which cover a range of from 1 in. of water to about 45 in. of mercury have been described.[3] The method of making the tests is also covered in the same bulletin, and, since it is standard hydraulic testing procedure, it will not be repeated here.

In general, the results of the testing have been relatively consistent for each orifice, and the computations which have all been checked, indicate that the results to date are as satisfactory as can be expected for the limited range of data.

Location of Vena Contracta

In all cases enough readings were taken along the downstream pipe section to determine experimentally the location of the vena contracta. It can be noted that the location of the vena contracta differs from that for concentric orifices. In all cases the vena contracta for eccentric and segmental orifices is farther downstream for the higher diameter ratios. For diameter ratios of less than 0.5, it is probable that the vena contracta is somewhat farther downstream than indicated in Fig. 1. However, the pressure curves below the orifices were so flat that it was difficult to determine exactly where they had their lowest value.

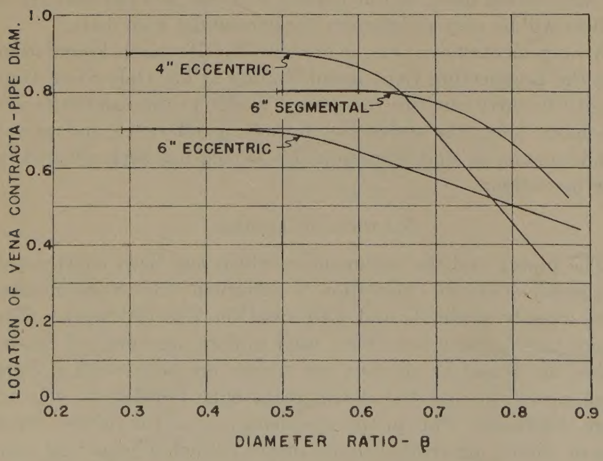

Fig. 1 Location of Vena Contracta for Segmental and Eccentric Orifices in 6-In. and 4-In. Pipe Lines (November 22, 1948.)

Method of Analysis

Fig. 2 shows the method of analysis used in plotting Figs. 3 to 8, inclusive. After the test runs were completed the results were calculated. A curve showing the variation of coefficient of discharge with $1/(\text{Reynolds number})^{1/2}$ was then plotted for each eccentric orifice. A sample curve is shown in the lower part of Fig. 2. A straight line was then drawn through the series of points with the slope indicated by the arrangement of the test points. A point read from the curve at a value of (1 over the square root of the Reynolds number) of 0.002 for each orifice was then plotted against diameter ratio, and a smooth curve drawn through these points. The slope of the straight line was also plotted against diameter ratio, and another smooth curve drawn through the slope points. A second line was then drawn on the original curve going through the point indicated on the smooth curve and with the slope indicated from the slope curve. This curve was then replotted as shown at the top of Fig. 2. In most cases the curves determined by the cross-plotting process fitted the points as closely as did the ones put in by eye.

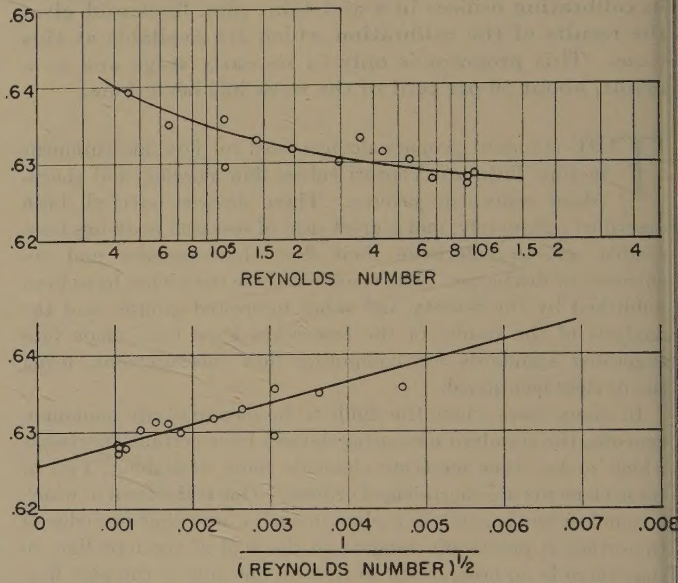

Fig. 2 Calibration of 4-In. Eccentric Orifice No. 80152; $\beta = 0.70$; Coefficient of Discharge Versus Reynolds Number (November 13, 1948.)

For the segmental orifices the test points were plotted as shown at the top of Fig. 2, and a curve drawn in, attempting to fit the points as closely as possible. These were then cross-plotted for several different values of Reynolds numbers and a smoothing process carried out.

The curves in Figs. 3 to 14, inclusive, are the final results of these cross-plotting methods.

Results

Figs. 13 and 14 are the summary curves for all of the orifices tested to date and include curves of concentric orifices (taken from Part 1 of the "Fluid Meters Report") for comparative purposes. Fig. 13 is for orifices with vena-contracta taps and Fig. 14 for orifices with flange taps. Both of these curves have been plotted for a fixed value of the Reynolds number at the orifice of 2.5×10^5. This value of Reynolds number was chosen because practically all of the orifices used had actual data at this Reynolds

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[3] "The Flow of Water Through Orifices," by S. R. Beitler, The Ohio State University Engineering Experiment Station Bulletin no. 89, May, 1935.

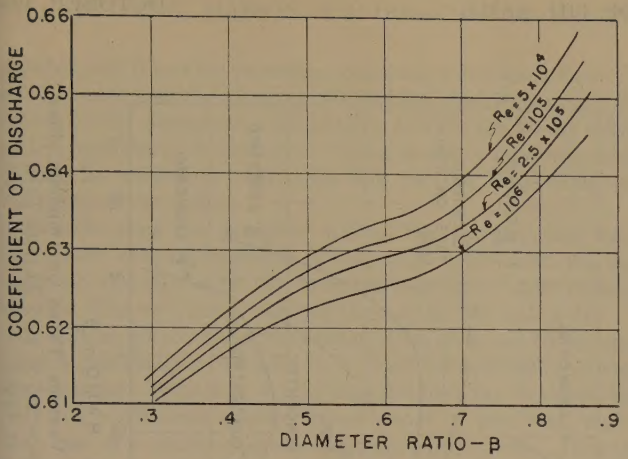

Fig. 3 Eccentric Orifices in a 4-In. Pipe Line; Coefficient of Discharge Versus Diameter Ratio; Vena-Contracta Taps
(November 12, 1948.)

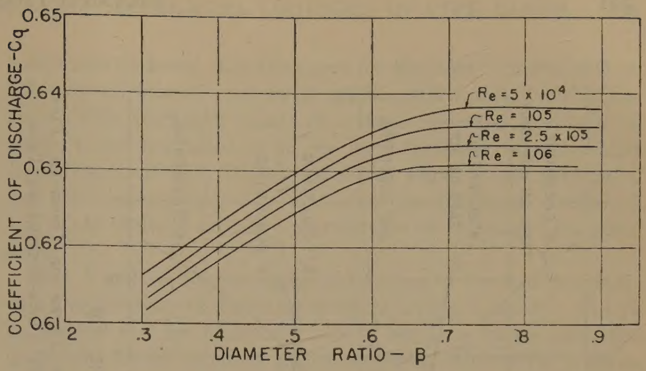

Fig. 4 Eccentric Orifice in a 4-In. Pipe Line; Coefficient of Discharge Versus Diameter Ratio; Flange Taps
(November 13, 1948.)

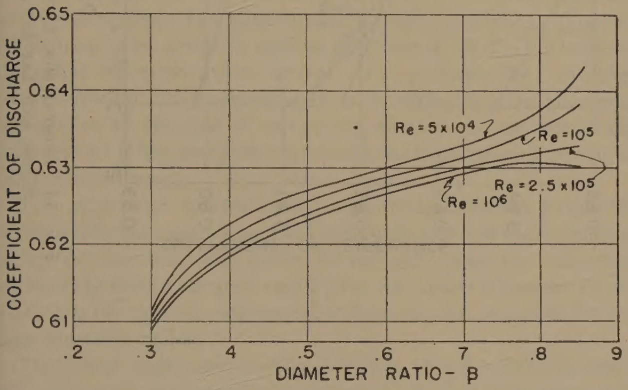

Fig. 5 Eccentric Orifices in a 4-In. Pipe Line; Coefficient of Discharge Versus Diameter Ratio; Vena-Contracta Taps at 90 Deg
(November 12, 1948.)

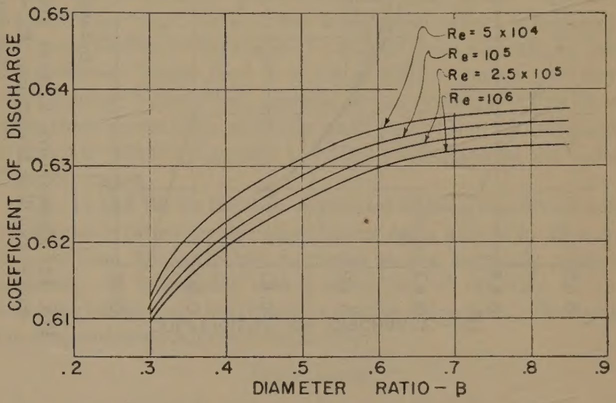

Fig. 6 Eccentric Orifices in a 4-In. Pipe Line; Coefficient of Discharge Versus Diameter Ratio; Flange Taps at 90 Deg
(November 12, 1948.)

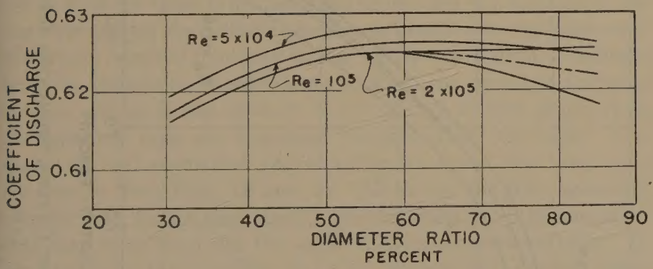

Fig. 7 Diameter Ratio—Coefficient Curves; Eccentric 6-In. Orifices; Vena-Contracta Taps

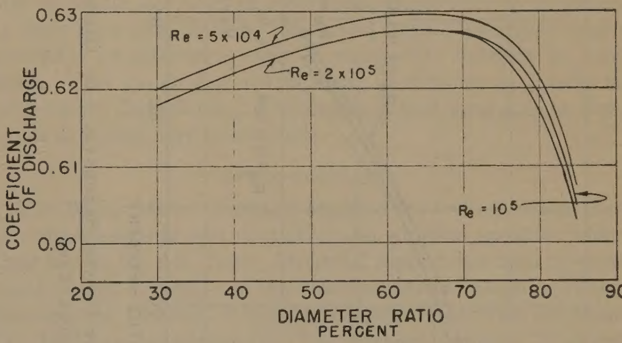

Fig. 8 Coefficient—Diameter Ratio Curves; Eccentric 6-In. Orifices; Flange Taps

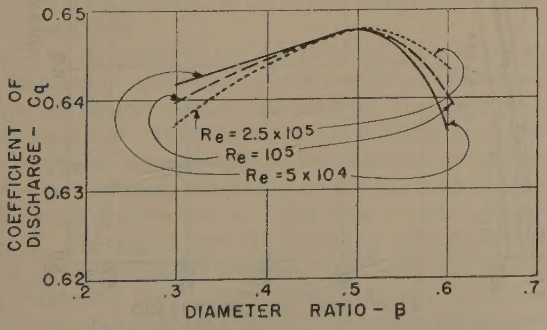

Fig. 9 Segmental Orifices in a 4-In. Pipe Line; Coefficient of Discharge Versus Diameter Ratio; Vena-Contracta Taps
(November 13, 1948.)

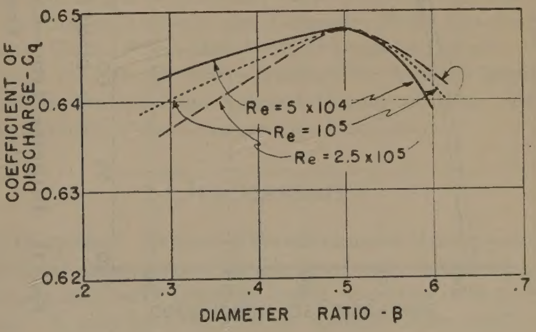

Fig. 10 Segmental Orifices in a 4-In. Pipe Line; Coefficient of Discharge Versus Diameter Ratio; Flange Taps
(November 13, 1948.)

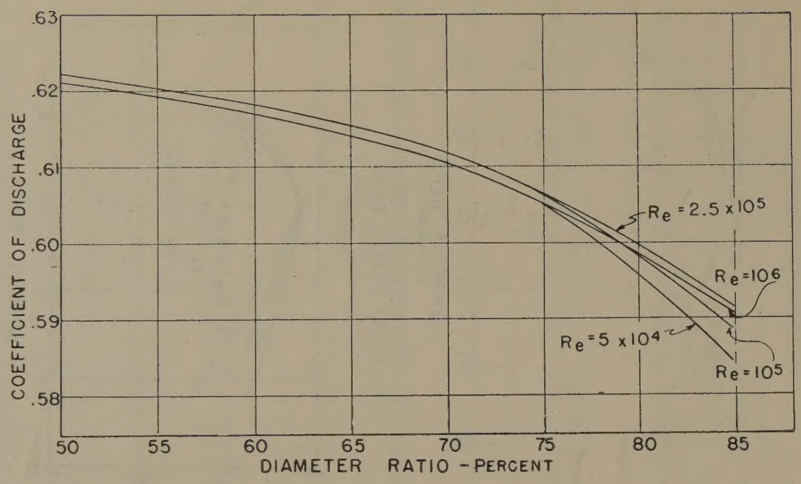

Fig. 11 Diameter Ratio—Coefficient Curves; Segmental 6-In Orifices; Vena-Contracta Taps

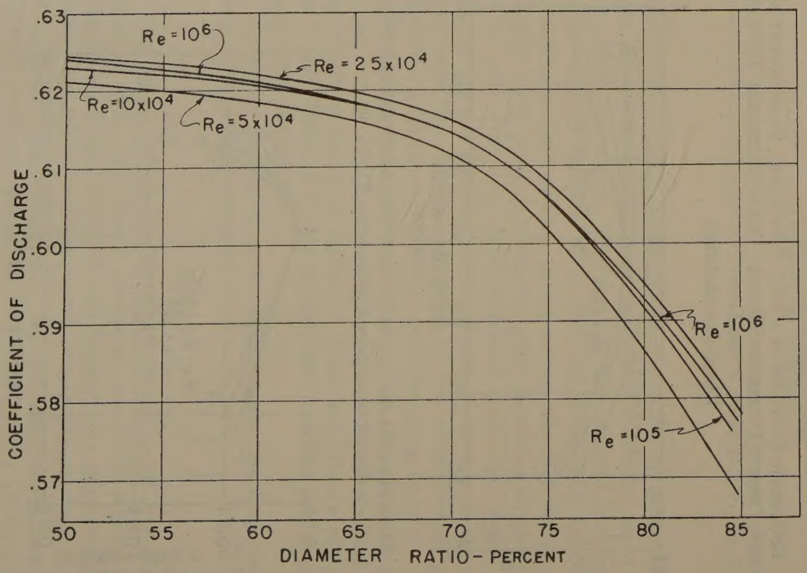

Fig. 12 Diameter Ratio—Coefficient Curves; Segmental 6-In. Orifices; Flange Taps

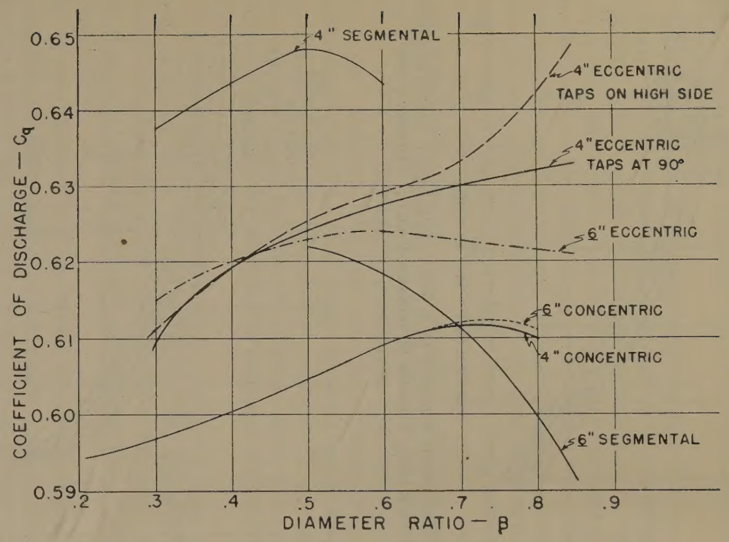

Fig. 13 Comparison Curves for Various Types of Orifices, Pipe Sizes, and Tap Locations; $Re = 2.5 \times 10^5$; Vena-Contracta Taps (November 13, 1948.)

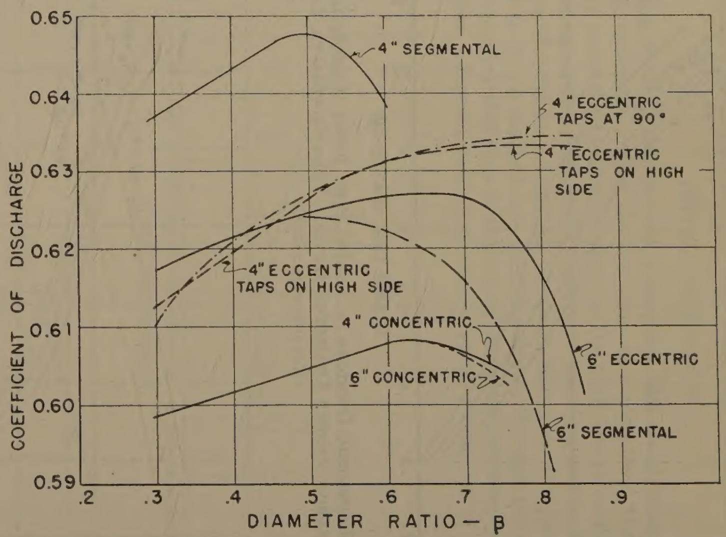

Fig. 14 Comparison Curves for Various Types of Orifices, Pipe Sizes, and Tap Locations; $Re = 2.5 \times 10^5$; Flange Taps (November 13, 1948.)

number, and it was not necessary to make any extrapolations. It will be noted that for diameter ratios of less than 70 per cent, the curves for the concentric orifices in Fig. 13 have the lowest values of the coefficients of discharge. There is also a relatively small change in coefficients with pipe size for the larger-diameter-ratio concentric orifices.

In calibrating the eccentric orifices, the 4-in-line tests have been made with the high side of the orifice in line with the taps and with the taps at 90 deg from the high side; while those in the 6-in. line were made with the taps on the high side only.

It will be noted that for diameter ratios of below 50 per cent there is practically no difference (i.e., less than 1 per cent) between the eccentric orifices in both pipe sizes and both tap locations, and that the coefficients appear to be about 3 per cent higher than that of a similar-diameter-ratio concentric orifice. For the larger diameter ratios, however, there is considerable difference between the 4-in. and the 6-in. eccentric-orifice curves; the coefficient values for the 4-in. orifices with vena-contracta taps on the high side continue to rise with diameter ratio, while the 6-in. eccentric-orifice coefficients fall slightly as the ratio increases. The coefficients of 4-in. eccentric orifices with taps at 90 deg fall between the other eccentric-orifice curves. It is probable that the high coefficients for the eccentric orifices is caused by the pipe wall being close to the side of the orifice, so that on one side of the orifice there is less reduction in area of the high-velocity stream at the vena contracta.

The segmental orifices show a somewhat different trend, the coefficient being about 8 per cent higher than that of the concentric orifice for the 4-in. line below 50 per cent diameter ratio, and apparently falling off very rapidly for the higher diameter ratios. The data for the 4-in. segmental orifices are not complete for the high diameter ratios.

The 6-inch segmental orifices were not tested below 0.5 diameter ratio and, as a result, no curve has been drawn below this point. It would appear from the data that it would fall parallel to the 4-in. curve, but about 5 per cent below it.

The curves for the flange taps, Fig. 14, have the same general arrangement as do the curves in Fig. 13. However, it will be noted that for these taps, the effect of rotating the location of the pressure tap on the eccentric orifices is practically negligible over the whole range of diameter ratios. This would seem to indicate that the difference in coefficients between the eccentric and concentric orifices is due entirely to the change in flow shape when the pressure taps are the same distance down the pipe. The eccentric and segmental curves differ by about the same percentage in both Figs. 13 and 14. However, the 6-in. eccentric orifices with flange taps show a greater drop in coefficient with diameter ratio than did the ones with vena-contracta taps. It would appear that possibly the 4-in. segmental curve will come tangent to the 6-in. segmental curve for the high diameter ratios. However, this cannot be predicted exactly, since this is an extrapolation, and the shape of the curve may be changed considerably by additional data.

Figs. 3 to 12, inclusive, are the individual plots of the various types and sizes of orifices in which the coefficient of discharge has been plotted against the diameter ratio for various values of the Reynolds number. It will be noted that in Fig. 3 the curves have about the same arrangement as for those for the concentric orifices, the coefficient decreasing as the Reynolds number increases for all diameter ratios. The same is true for Figs. 4, 5, 6, and 7. For Fig. 8 the 6-in. eccentric orifices with flange taps for diameter ratios of less than 70 per cent, the dependence of the coefficients on Reynolds numbers is very small, and for Reynolds-number values over 2×10^5 the coefficients appear to be practically constant. In Fig. 7 it will be noted that the same thing is true for the vena-contracta taps with the 6-in. orifices, and it will also be noted that the curve for Reynolds number of 2×10^5 is drawn with two values for all diameter ratios above 60 per cent. This particular series of orifices indicated a spread of about 1 per cent total at 85 per cent diameter ratio and high Reynolds numbers. The dotted curve which is the average of the other curves probably represents the predicted coefficient within the limits of accuracy of prediction for this high a diameter ratio.

Figs. 9 and 10, for the segmental orifices in the 4-in. line, indicate the normally expected trend for this type of orifice. It will be noticed that for diameter ratios of less than 50 per cent, the coefficient varies inversely as the Reynolds number for a given diameter ratio, while for over 50 per cent the slope in the curves is reversed and they vary directly as the Reynolds number. This tendency has been noticed in many cases with segmental orifice shapes and is probably due to the fact that as the area ratio increases, the shape of the stream leaving the orifice becomes more and more similar to the shape of a stream from a nozzle, so that the coefficient would tend to increase with an increase of Reynolds number rather than decrease as an ordinary orifice does. Since these curves are a relatively small part of the ordinary range of use, it is difficult to predict how they will behave for the higher diameter ratios.

Figs. 11 and 12, for the 6-in. segmental orifices, are plotted only for diameter ratios greater than 50 per cent, and so in all cases indicate that the coefficient increases as the Reynolds number increases. It is possible that a reanalysis of the data on the 6-in. size based upon the complete results of the 4-in. will change the shape of these curves considerably.

Conclusion

In conclusion, it should be reiterated that these curves are probably different in shape and absolute value from the curves which ultimately will be given for this particular type of orifice. However, it is believed that coefficients taken from these curves will be within the range of commercial reproducibility of orifices in these pipe sizes, diameter ratios, and Reynolds-number ranges, providing no attempt is made to extrapolate the data to other sizes. This material is of sufficient importance to justify its preliminary dissemination at this time, so that some data on these types of orifices may be available.

Acknowledgment

The Bailey Meter Company supplied the orifices at less than the manufacturing cost, and they also supplied the data which they had on the 6-in. lines. Financial support has been received (as of August 1, 1949) from the American Meter Company, Brown Instruments Division of the Minneapolis-Honeywell Regulator Co., Builders-Providence, Inc., The Foxboro Company, Rockwell Manufacturing Company, Republic Flow Meters Co., Standard Oil Development Co., Subsidiary Companies of Bethlehem Steel Corporation, Hagan Corporation, Taylor Instrument Companies, E. I. du Pont de Nemours and Company, Shell Development Company, United States Steel Corp. of Delaware, Simplex Valve and Meter Co., Daniel Orifice Fitting Co., Askania Regulator Co., and the Technical and Research Committee of the Natural Gas Department of the American Gas Association.

Discussion

A. P. Colburn.[4] Because of the convenience of using eccentric or segmental orifices where liquids may carry entrained air or where gases may carry entrained liquid, these new data are very welcome. It may be of interest to compare these results with

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[4] Assistant to the President, University of Delaware, Newark, Del. Mem. ASME.

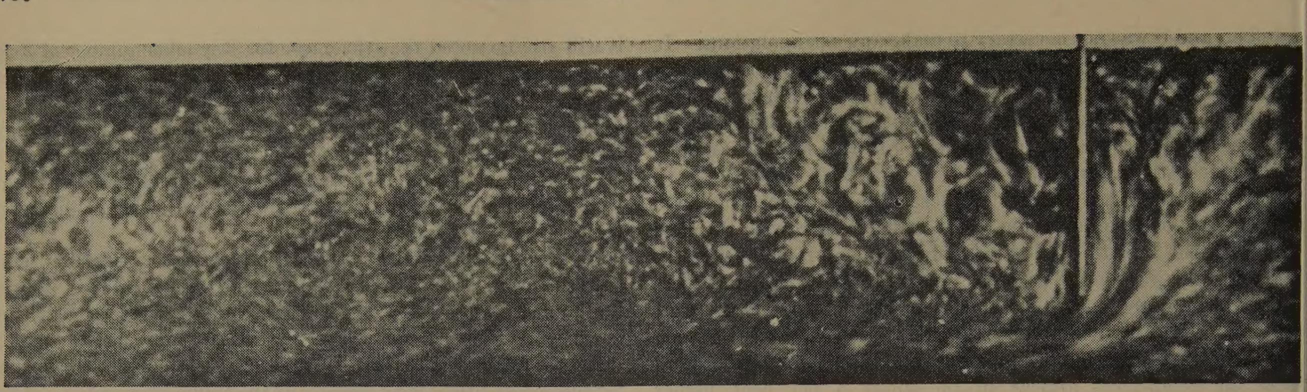

Fig. 16 Type of Flow Occurring Through an Eccentric Orifice

(Conditions: Slurry of aluminum powder and water flowing in a channel 1 in. deep covered by a glass plate and photographed by reflected light.)

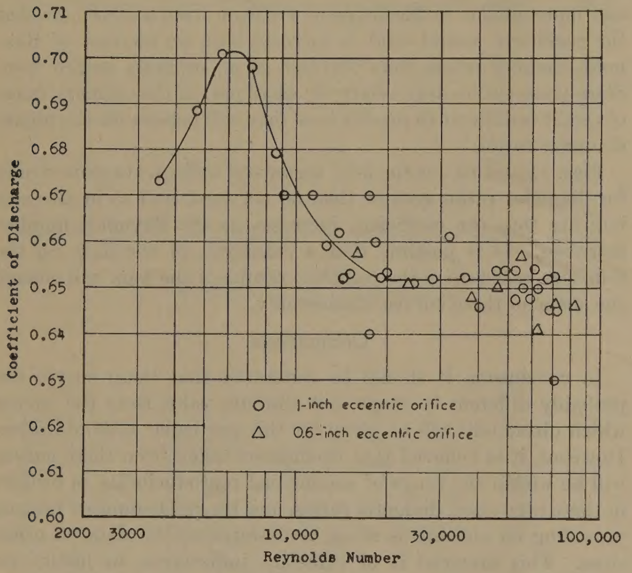

Fig. 15 Coefficient of Discharge Versus Reynolds Number for Sharp-Edged Eccentric Orifices Studied

(Diameter of pipe = 2-in. standard.)

some obtained at the University of Delaware.[5] These results may be summarized as follows:

1 Data were obtained for flow of water in a 2-in. steel pipe with eccentric orifices of 1.0 and 0.6 in. diam, over a range of Reynolds numbers (based on orifice diameter) from 4000 to 80,000. Above 20,000, the values of orifice coefficients (corrected for velocity of approach) averaged 0.65, compared with the usual value of around 0.62 for concentric orifices of the same orifice-to-pipe diameter ratios. Data at these and other Reynolds numbers are shown in Fig. 15, herewith. The orifices used were made of $^1/_8$-in. sheet steel, beveled 60 deg to a depth of $^3/_{32}$ in. on the downstream side, and were located just tangent to the inside wall of the pipe.

2 No difference in reading was observed within a range of downstream manometer tap locations of from 0.7 to 1.5 pipe diam. The upstream tap was about 1 pipe diam from the upstream face of the orifice. The taps were located in the bottom of the pipe, thus being opposite from the eccentric-orifice location.

3 Visual studies in a special two-dimensional rig supported these results qualitatively by indicating less contraction than for concentric orifices and a wider range of nearly constant contraction, as can be seen from the view in Fig. 16 of this discussion.

---

[5] "Fundamental Studies of Eccentric Orifices," by E. B. Taylor, B.Ch.E. thesis, University of Delware, 1942.

Critical Flow Through Sharp-Edged Orifices[1]

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An investigation was made to discover the flow rates of air through sharp-edged orifices in which the pressure drop was below the critical; the relationship of pressure, temperature, and orifice area; and a comparison between the nozzle and the orifice. Equations are derived for expressing orifice flow in both the critical and subcritical regions.

NOMENCLATURE

The following nomenclature is used in the paper:

A = orifice area, sq in.
a = orifice area, sq ft
C_m = discharge coefficient (as applied to equation given by Bean, Buckingham, and Murphy)
d = orifice diameter, in.
g = acceleration of gravity; 32.17 ft per sec per sec
n = ratio of specific heats; 1.4 for air
K = coefficient of discharge
K_c = coefficient of discharge
P_1 = pressure, upstream to test orifice, psia
P_2 = pressure, downstream to test orifice, psia
p = pressure, psf
Q = volumetric flow, cfm
R = gas constant; 53.3 for air
R_e = Reynolds number
r = pressure ratio, P_2/P_1
T = temperature ahead of test orifice, deg R
V = velocity, fps
Y = expansion factor
β = ratio of orifice diameter to pipe diameter
ρ = density, lb per cu ft
$\Omega = \dfrac{w\sqrt{T}}{AP_1}$, $(T)^{1/2}$/sec
$\Xi = \dfrac{w\sqrt{T}}{A}$, $\dfrac{\text{lb } (T)^{1/2}}{\text{sec in.}}$

INTRODUCTION

Critical flow is defined as the flow of a compressible fluid through a distance during which the pressure drops below the critical pressure. The critical pressure of a compressible fluid may be defined as the minimum pressure which can be obtained in a fluid stream at the smallest cross section of the passage (throat) through which the fluid is flowing for any given upstream pressure. The ratio of this minimum pressure in the throat to the upstream pressure is practically constant for any given fluid. It has been found to be 0.49 for monotomic gases, 0.53 for diatomic gases, and slightly higher for gases of greater atomic complexity. When the cond tions of critical flow have been reached, it can be shown that the velocity at the throat is equal to the velocity of sound at the temperature in the throat. Hence for air, which was the only gas with which the author experimented, critical flow is flow during which the pressure ratio r, or the ratio of the downstream pressure to the upstream pressure, varies between $r = 0.53$ and $r = 0$. Subcritical flow is flow during which the pressure ratio varies between $r = 1.0$ and $r = 0.53$.

Because of the fact that for a given upstream pressure, the throat pressure cannot change at pressure ratios less than the critical, it is found that for the case of an ideal nozzle, the mass flow reaches a maximum after which no decrease in pressure beyond the throat can alter the flow. Actual nozzles have been investigated which follow ideal flow within 1 per cent.

It has long been known that the flow of a gas through a sharp-edged orifice (also known as square-edged, thin-plate, etc.) does not follow the ideal flow law as does flow through a nozzle. Perhaps the most remarkable difference is found in the critical-flow range where investigations of the author have shown that the flow not only is not constant at pressure ratios below the critical but actually increases to a maximum of about 13.7 per cent. This phenomenon is due to the fact that, unlike a nozzle, the efflux is unconfined. Thus, while the velocities are those obtained by an adiabatic expansion, the area of the jet is free to change. Stanton (1)[3] showed that as the pressure surrounding the free jet decreases, the cross-sectional area of the jet increases. In a liquid the area of the jet is essentially a constant.

The purpose of this paper is not only to supply information about the critical-flow range, but also to supplement existing data which have been published about flow in the subcritical-flow range. The ASME Research Committee on Fluid Meters has published detailed reports concerning subcritical flow (2). Probably because of the limited demand, all coefficients and expansion factors relate to diameter ratios β, of 0.1 and above. The author investigated orifices of diameter ratios as low as 0.0757.

TEST METHODS

The apparatus was set up with two orifices in series as shown in Fig. 1. The first orifice was used for measuring the flow rate, and the second orifice was the test orifice. As was previously stated, the subcritical range of flow has been investigated thoroughly; and by following the instructions laid down by the ASME (2), it is possible to construct a measuring orifice whose accuracy is within the limits desired for this test.

As it was desired to carry out all tests with a negligible velocity of approach, the first step was to select the diameter of the chamber preceding the test orifice. It was decided that a $^1/_4$-in. orifice was the largest size that would accomodate the capacity of the compressor available and still maintain a head pressure on the test orifice of over 100 psia. By using 2-in. pipe (actual internal diameter equal to 2.067 in.), the diameter ratio β, would be 0.1209 and the velocity of approach factor

$$\sqrt{\frac{1}{1-\beta^4}}$$

would be about 1.0001, or, for purposes of the experiment, negli-

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NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society. Paper No. 48—A-146.

[3] Numbers in parentheses refer to the Bibliography at the end of the paper.

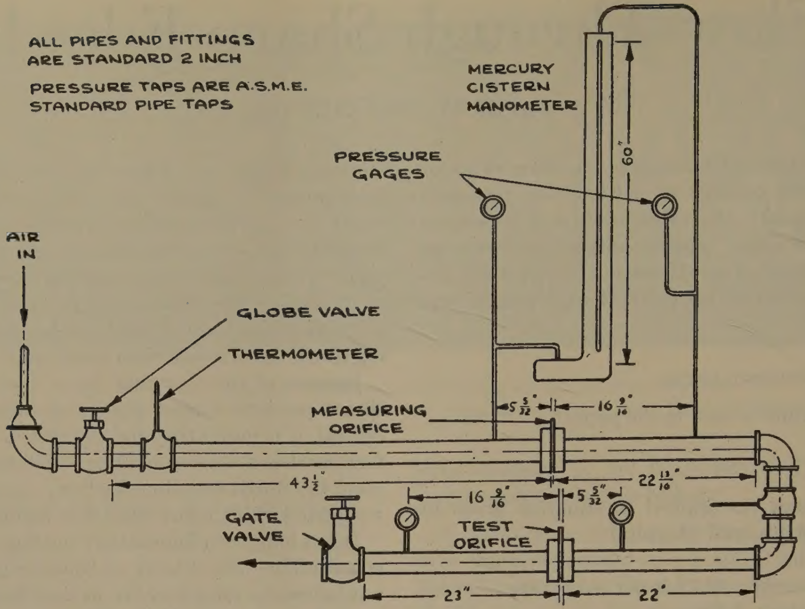

Fig. 1 Piping Arrangement Used for Measuring Flow Through Sharp-Edged Orifices

gible. Therefore 2-in. pipe was thought to be entirely satisfactory.

The next step was to select the position of the pressure taps. Pipe taps[4] were chosen because they give the permanent pressure loss. This way it might be possible to apply the results to a very large chamber separated by a plate containing a small orifice, or perhaps even to a theoretical analysis where it might be desirable to assume an upstream and downstream chamber of infinite diameter. Actually, the question of taps becomes of minor importance when working with small diameter ratios. The flowmeter handbook (3) indicates that for $\beta = 0.1$ the difference in flow for two setups, one with pipe taps and the other with flange taps or vena-contracta taps, all reading the same differential pressure, would be about $^1/_2$ per cent.

The orifice plates were made from stainless steel. The holes were drilled and reamed. The excess material on the downstream side of the hole was removed with a large drill. The right angle on the upstream face was obtained by hand polishing with a very fine stone. The finished plates were measured with plug gages. A drawing showing the method of construction of the orifice plates appears in Fig. 2.

Table 1 gives the orifices which were used for the tests.

TABLE 1 ORIFICES USED IN TESTS

Identification letter	Diameter, in.	Diameter ratio
F	0.2500	0.1209
J	0.2186	0.1058
L	0.2040	0.0986
M	0.1877	0.0908
Q	0.1564	0.0757

All gages used were quality test gages. These gages were checked on dead-weight gage testers at least once a day during the tests.

Two mercury thermometers were used to measure the temperatures ahead of each orifice (see Fig. 1).

A 60-in. cistern-type mercury manometer was used to measure the differential pressure across the measuring orifice.

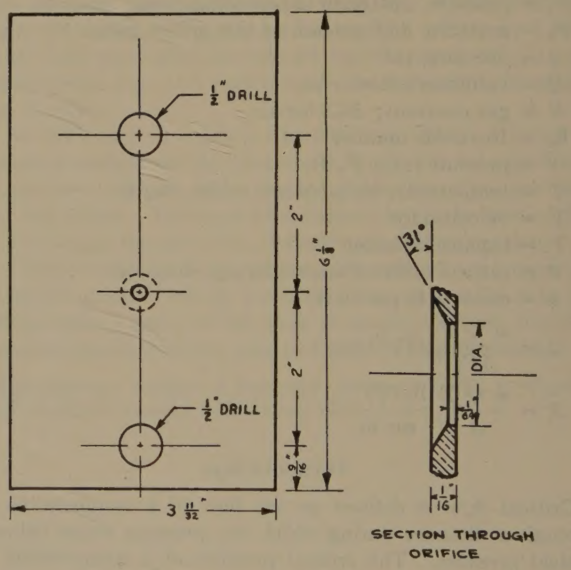

Fig. 2 Method of Constructing Orifice Plates

A reciprocating-steam-engine-driven air compressor with a capacity of 50 cfm supplied the high-pressure air. This discharged directly into a large storage tank, and it was possible to obtain steady pressures for long periods of time. All air was used at the temperature of the compressor discharge which varied between 80 and 100 F. No arrangements were made to vary the air temperature. No correction was made for humidity.

The tests were run as follows: The orifices were placed in series as shown in Fig. 1. By means of regulating the valves, the desired pressures above and below the test orifice, P_1 and P_2, were obtained. These were recorded and immediately afterward the pressure and temperature above the measuring orifice, the differential pressure across the measuring orifice, and the temperature ahead of the test orifice, were recorded. By using the methods outlined by the ASME (2), the flow rate through the measuring orifice was determined and applied to the test orifice.

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[4] Pipe taps are positioned so that one tap is $2^1/_2$ pipe diam from the upstream face and the other 8 diam from the downstream face.

TEST RESULTS

To begin a description of the tests, it will be necessary to explain the two flow factors that are used by the author, Ξ and Ω. Flow through an orifice is found to be a function of the flow area, the Reynolds number, the diameter ratio, the pipe diameter, the specific-heat ratio, the head pressure, the back pressure, and the absolute temperature; thus

$$w = f(A, \mathbf{R}_e, \beta, D_p, n, P_1, P_2, T)$$

The effect of the Reynolds number tends to become constant above $\mathbf{R}_e = 100{,}000$, and all tests were run abovethis number; it will be shown that at very small diameter ratios the effect of the diameter ratio and the size of pipe tend to become negligible. In almost every theoretical and empirical equation of gas flow, the flow is found to vary inversely as the square root of the absolute temperature. That assumption is made here. That the flow varies directly with the area will be shown later. So if the foregoing is true

$$w = \frac{A}{\sqrt{T}}\, f(P_1, P_2)$$

Then, grouping w, A, and T gives

$$\frac{w\sqrt{T}}{A} = f(P_1, P_2)$$

This factor will be called Ξ. So $\Xi = w\sqrt{T}/A$, and Ξ is seen to be a function of the absolute pressures.

Now suppose Ξ is divided by P_1. Then

$$\frac{\Xi}{P_1} = \frac{w\sqrt{T}}{P_1 A}$$

This factor will be called Ω, and $\Omega = (w\sqrt{T})/(P_1A)$. Ω is seen to be a function of the pressure ratio alone.

The significance of Ω will be seen more clearly by considering the equation representing an adiabatic expansion through an ideal nozzle

$$w = \frac{P_1 A}{\sqrt{T}} \sqrt{\frac{2g}{R}\,\frac{n}{n-1}}\; r^{1/n} \sqrt{1 - r^{n-1/n}} \ldots\ldots [1]$$

Subsequently

$$\Omega = \sqrt{\frac{2g}{R}\,\frac{n}{n-1}}\; r^{1/n} \sqrt{1 - r^{n-1/n}} \ldots\ldots\ldots [1a]$$

All tests were run holding a constant head pressure and varying the back pressure. Values of Ξ were calculated for each run. A plot of Ξ versus P_2 appears in Fig. 3.

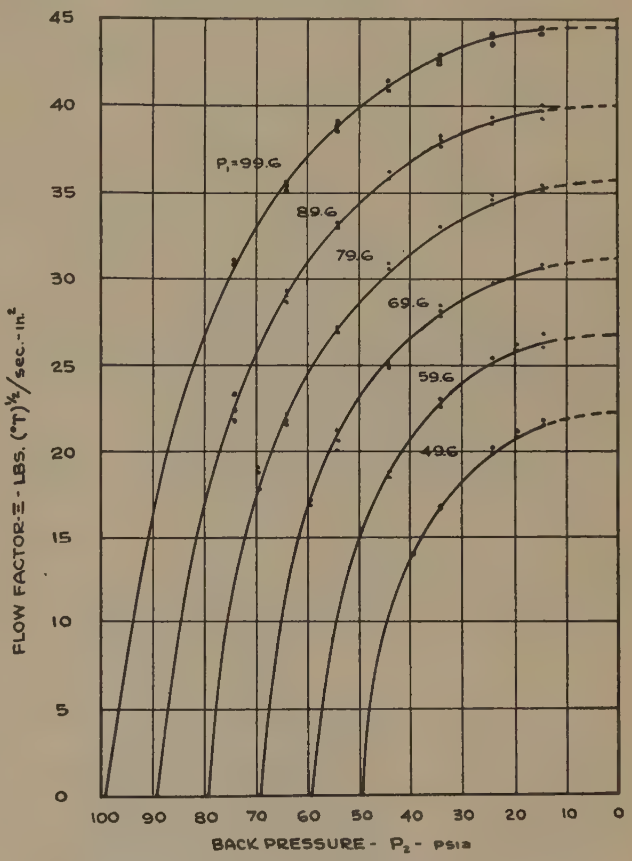

FIG. 3 FLOW THROUGH SHARP-EDGED ORIFICES AT CONSTANT HEAD PRESSURE

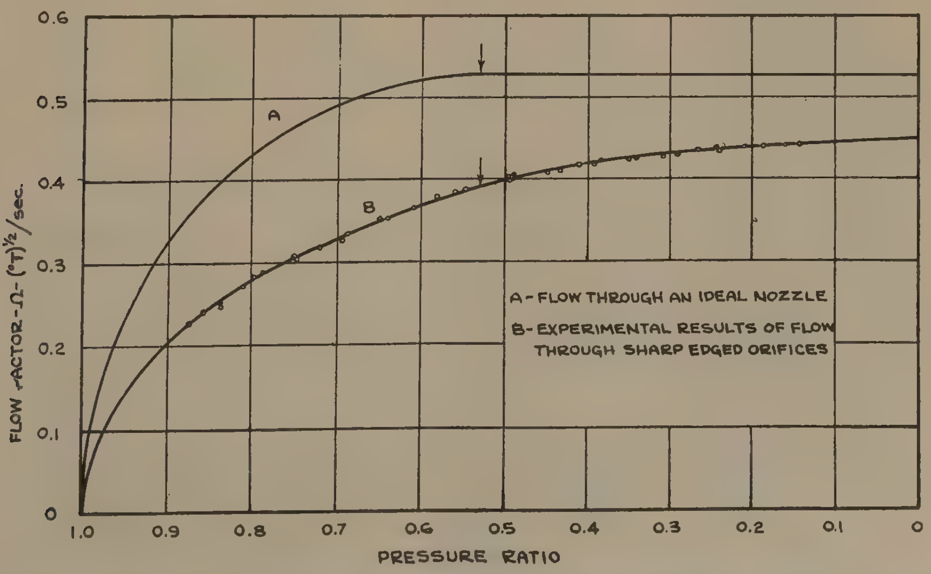

FIG. 4 FLOW THROUGH SHARP-EDGED ORIFICES COMPARED TO FLOW THROUGH AN IDEAL NOZZLE

Of themselves these curves are of little value. However, the next step was to divide each of these curves by the parameter, P_1. The interesting result is that all of these curves now seem to lie on the same line as is seen in Fig. 4. This is a plot of Ω versus r. The values of Ω were obtained from the "smoothed" data taken from Fig. 3. As in the nozzle flow equation, Ω appears to be a function of r only.

As will be noted from Fig. 4, it would be rather difficult to draw an accurate line through the lower-pressure-ratio section because of the flatness of the curve. To study the characteristics of orifice flow more closely Ω versus r was plotted on elliptical co-ordinates, i.e., all values of Ω and r were squared.

The orifice characteristic is shown on elliptical co-ordinates in Fig. 5. Some interesting conclusions can be drawn from this plot. First it is seen that there is a fundamental difference in the nature of the flow in the subcritical region and in the critical region. In the subcritical region the points are found to lie on a straight line; and this of course represents an ellipse. In the critical region, the flow characteristic falls away from an ellipse and appears to be an entirely different function. It is interesting to note that although the lowest pressure ratio obtained by test was 0.146, on elliptical co-ordinates this point falls so close to the zero axis that the delicate question of extrapolating into the region of very low pressure ratios is solved with a minimum of doubt.

It is now a simple matter to express the flow relationship in the subcritical region by an equation of an ellipse

$$\Omega = 0.465\sqrt{1 - r^2} \qquad [2]$$

Or in terms of mass flow

$$w = 0.465\,\frac{A}{\sqrt{T}}\,\sqrt{P_1{}^2 - P_2{}^2} \qquad [3]$$

Because the characteristic in the critical region is obviously not elliptical, it was necessary to use another approach in this region. Subsequent discussion will explain the development of a formula for orifice flow in the critical region and also present a correlation between this work and that of the Society.

The fundamental equation which expresses velocity of an incompressible fluid in terms of its equivalent pressure is

$$\Delta p = \frac{\rho V^2}{2g} \qquad [4]$$

or

$$V = \sqrt{2g\,\frac{\Delta p}{\rho}} \qquad [5]$$

but

$$w = \rho a V$$

so substituting gives the relationship of mass flow through an opening of area, a, caused by a pressure differential

$$w = \rho a\sqrt{2g\,\frac{\Delta p}{\rho}}$$

$$= a\sqrt{2g\rho\,\Delta p} \qquad [6]$$

or, finally, changing pressure to pounds per square inch and area to square inches

$$w = 0.668\,A\sqrt{\rho\,\Delta P} \qquad [7]$$

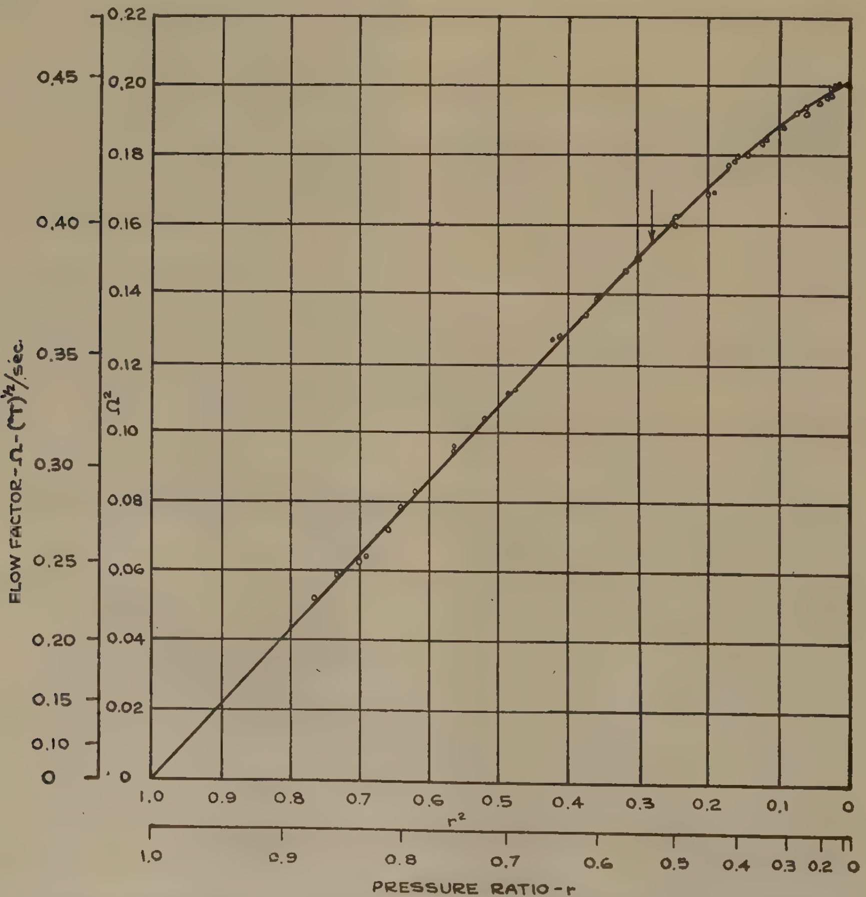

Fig. 5 Flow Through Sharp-Edged Orifices Plotted on Elliptical Co-Ordinates

The ASME (2) makes three corrections to this formula. The first, the coefficient of discharge K, combines the coefficient of contraction, the coefficient of velocity, and the velocity of approach factor. The correction E, is a correction for expansion of the orifice due to high operating temperatures. Since no extreme temperature was used for the tests, this factor may be omitted. The third correction is the expansion factor, Y. This factor permits the formula for incompressible fluids to be used with compressible fluids. Y is the ratio between the flow of two fluids, everything being equal, except that one is compressible and the other is incompressible. So the final form is

$$w = 0.668\, AKEY \sqrt{\rho \Delta P} \ldots\ldots\ldots\ldots\ldots [8]$$

Omitting E

$$w = 0.668\, AKY \sqrt{\rho \Delta P} \ldots\ldots\ldots\ldots\ldots [9]$$

Noting that for air $\rho = 2.70P/T$ and that $\Delta P = P_1(1 - r)$ this may be rewritten

$$w = 1.096 \frac{P_1 A}{\sqrt{T}} KY \sqrt{1 - r} \ldots\ldots\ldots\ldots [10]$$

And expressing this in terms of Ω

$$\Omega = 1.096\, KY \sqrt{1 - r} \ldots\ldots\ldots\ldots\ldots [11]$$

Rearranging terms

$$KY = \frac{\Omega}{1.096 \sqrt{1 - r}} \ldots\ldots\ldots\ldots\ldots [12]$$

Now, by inserting values of Ω as determined by the tests, the relation between KY and the pressure ratio may be investigated. This relation is shown graphically in Fig. 6. It is interesting to note the decisive change in slope occurring in the region of the critical ratio.

Now KY may also be expressed by means of equations. For the subcritical region, substituting $\Omega = 0.465 \sqrt{1 - r^2}$ in Equation [12] gives

$$KY = 0.425 \sqrt{1 + r} \ldots\ldots\ldots\ldots\ldots [13]$$

It is seen that the line has a very slight curvature.

In the critical region the KY-function appears to be a straight line, and by graphical methods it was found that it could be expressed

$$KY = 0.410 + 0.220r \ldots\ldots\ldots\ldots\ldots [14]$$

By using this last expression for KY, it will now be shown how an expression for critical flow may be developed. Substituting Equation [14] into the ASME flow equation, Equation [9], results in

$$w = 0.668A(0.410 + 0.220r) \sqrt{\rho \Delta P} \ldots\ldots\ldots [15]$$

or

$$w = \frac{AP_1}{\sqrt{T}} (0.449 + 0.241r) \sqrt{1 - r} \ldots\ldots\ldots [16]$$

or in terms of Ω

$$\Omega = (0.449 + 0.241r) \sqrt{1 - r} \ldots\ldots\ldots [17]$$

So now two equations, Equations [3] and [16], have been developed which mathematically describe the behavior of air flowing through a sharp-edged orifice. The limitations to these expressions should be emphasized. They apply only to air. They apply only to orifices with a negligible velocity of approach. They apply only to the pressure and temperature conditions heretofore prescribed.

Now, returning to the function KY, an interesting speculation may be made. The fact that in the critical region the values of KY plot on a straight line has an important significance. At high rates of flow the coefficient of discharge K tends to become constant. Therefore, if K were a constant, Y would be a linear function of the pressure ratio. In the subcritical region the ASME (4) gives the expansion factor as

$$Y = 1 - [0.330 + 1.145(\beta^2 + 0.7\beta^5 + 12\beta^{13})] \frac{1 - r}{n} \ldots [18]$$

So then, for any given value of β, Y is a linear function of the pressure ratio. This equation holds up to the critical-pressure ratio. So, considering the fact that the expansion factor appears to be a linear function in both the subcritical and critical regions,

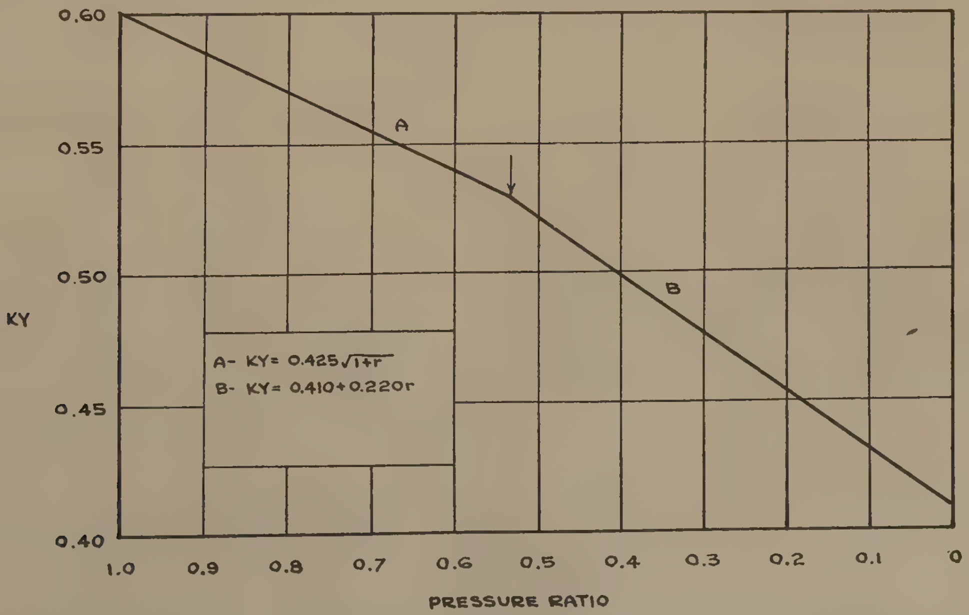

FIG. 6 VARIATION OF PRODUCT OF COEFFICIENT OF DISCHARGE AND EXPANSION FACTOR WITH PRESSURE RATIO

it would seem that the difference might lie only in the slopes. So it appears that if the original tests of Bean, Buckingham, and Murphy (5) had been carried down into the critical region, it might be possible to calculate flows at any pressure ratio simply by supplying an additional set of expansion factors.

Referring back to Fig. 4, curve A is seen to represent ideal nozzle flow (as expressed by Equation [1a]). The ratio of the vertical height of the orifice curve to the vertical height of the nozzle curve at any pressure ratio represents the flow coefficient K_c, or the fraction of flow through an orifice compared to that through an ideal nozzle under the same conditions of temperature and pressure, and both having the same area. The flow coefficient throughout the entire range is shown in Fig. 7. This curve also has another significance. If the velocity of the air at the vena contracta may be taken to be the velocity developed in an adiabatic expansion, then the velocities in the orifice and in the nozzle will be identical for any pressure ratio, and ratios of the mass flows will vary according to the curve in Fig. 7. So then, if the velocities and densities are the same, the curve in Fig. 7 will also represent the ratio of the area of the vena contracta to the area of the orifice.[5]

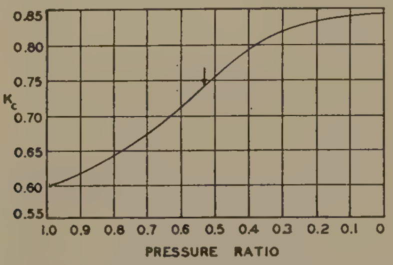

Fig. 7 Flow Coefficient for Sharp-Edged Orifice With Zero Approach Velocity

By applying K_e to the theoretical flow formula, Equation [1], it would seem reasonable to assume that flow of any gas through an orifice could be calculated. In the critical region, Equation [1] reaches a maximum value which remains constant at pressure ratios below the critical

$$w = 0.259 \frac{P_1 A}{\sqrt{T}} \sqrt{\frac{2g}{R} \frac{n}{n-1}} \ldots\ldots\ldots\ldots [19]$$

And for air at moderate temperatures

$$w = 0.532 \frac{P_1 A}{\sqrt{T}} \ldots\ldots\ldots\ldots\ldots\ldots [20]$$

Comparison With Other Investigations

Bean, Buckingham, and Murphy (7) found that the use of a mean density ρ_M, or the density of the air at the upstream temperature and at the average of the upstream and downstream pressures in the simple hydraulic equation, resulted in a coefficient of discharge which varied only a few per cent from the coefficient for water over the whole range from $r = 1$ to $r = 0.5$. The coefficient was defined by

$$w = C_M \frac{0.525\, d^2}{\sqrt{1 - \beta^4}} \sqrt{\rho_M \Delta P} \ldots\ldots\ldots\ldots [21]$$

and for air

$$\rho_M = \frac{2.70}{T} \left[\frac{P_1 + P_2}{2} \right] = 1.35 \frac{P_1 + P_2}{T}$$

So, substituting this value of ρ_M, and letting $\beta \rightarrow 0$ results in

$$w = C_M\, 0.776 \frac{A}{\sqrt{T}} \sqrt{P_1^2 - P_2^2} \ldots\ldots\ldots [22]$$

Now, if C_M is equal to 0.60, which it very nearly is from $r = 0.9$ to $r = 0.7$, Equation [22] becomes

$$w = 0.465 \frac{A}{\sqrt{T}} \sqrt{P_1^2 - P_2^2} \ldots\ldots\ldots\ldots [23]$$

which is identical to Equation [3]. So for this range of flow the author's work agrees almost perfectly with that of Bean, Buckingham, and Murphy. However, C_M falls off in the lower pressure ratios incurring a maximum variation of $1^1/_2$ per cent at $r = 0.5$.

Fig. 8 represents a comparison of the author's results to an investigation made by Cox and Germano (8) of the work of Stanton (9), and Schiller (10). This figure shows the percentage increase of flow of a gas through an orifice over the flow at the critical-pressure ratio.

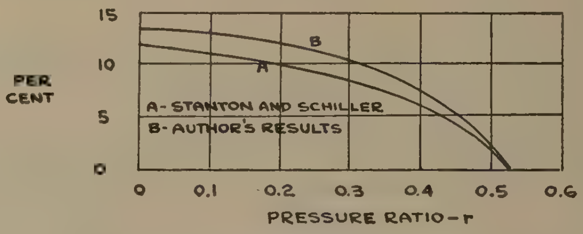

Fig. 8 Percentage Increase in Flow Over Flow at Critical-Pressure Ratio

One of the few people to attempt to describe orifice flow throughout the entire range of pressure ratios was H. B. Reynolds (11). Reynolds expresses orifice flow by

$$Q = 405 \frac{A}{\sqrt{T}} (P_1^2 - P_2^2)^{0.48} \text{ cfm at 32 F} \ldots\ldots\ldots [24]$$

This also may be expressed

$$w = 0.546 \frac{A}{\sqrt{T}} (P_1^2 - P_2^2)^{0.48} \text{ lb per sec} \ldots\ldots [25]$$

or

$$\Xi = 0.546 (P_1^2 - P_2^2)^{0.48} \ldots\ldots\ldots\ldots\ldots [26a]$$

$$= 0.546 P_1^{0.96} (1 - r^2)^{0.48} \ldots\ldots\ldots\ldots [26b]$$

It would not be possible to compare his results with those of the author except at a specific pressure. This was done in Table 2 for a head pressure of 100 psia. It is seen that at low and high pressure ratios Reynolds equation gives higher values than those of the author, while at intermediate values the author's values are higher. It might be well to mention that Reynolds used 2-in. pipe, but he placed one tap 6 in. ahead of the orifice and one directly behind it.

Since the test apparatus was constructed according to ASME Specifications, it is possible to compare the author's results with those of the ASME in the subcritical region for $\beta = 0.1$. Values of Ω were calculated by Equation [11] at $\beta = 0.1$. The coefficient of discharge is nearly constant, and was taken at the values given in Table 3. A direct comparison can be made for

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[5] An investigation by Stanton (6) indicates that the vena contracta occurs at the position in the jet where the velocity reaches the adiabatic velocity in a subcritical expansion and where the velocity reaches the sonic velocity in a critical expansion.

TABLE 2 COMPARISON OF RESULTS TO THOSE OF H. B. REYNOLDS

(P_1 = 100 psia)

Pressure ratio	Ω	
	Reynolds	Author
1.0	0	0
0.95	10.74	10.35
0.9	20.4	20.2
0.8	27.8	27.9
0.7	32.9	33.2
0.6	36.6	37.2
0.5	39.6	40.3
0.4	41.5	42.3
0.3	43.4	43.6
0.2	44.5	44.4
0.1	45.2	44.8
0	45.4	44.9

two-inch pipe, as the author's data were taken entirely on two-inch pipe; however, a comparison is also shown for 6 in. and 12 in. pipe for β = 0.1.

TABLE 3 COMPARISON OF RESULTS TO FLOW RATES OBTAINED BY MEANS OF THE ASME METHOD OF CALCULATION

	Values of Ω			
		Pipe size		
r	Author	2 in. K = 0.607	6 in. K = 0.601	12 in. K = 0.600
0.9	0.203	0.205	0.203	0.2025
0.8	0.279	0.282	0.280	0.278
0.7	0.332	0.3375	0.334	0.333
0.6	0.373	0.378	0.374	0.373
0.5	0.403	0.412	0.408	0.407

Strangely enough, the author's results agree more closely with the larger sizes of pipe. Actually, the value at r = 0.5 is not a valid comparison because this value is not guaranteed by the ASME.

CONCLUSIONS

1 Flow of air through a sharp-edged orifice with a negligible velocity of approach may be expressed in the subcritical region by

$$w = 0.465 \frac{P_1A}{\sqrt{T}} \sqrt{1 - r^2}$$

and in the critical region by

$$w = \frac{P_1A}{\sqrt{T}} (0.449 + 0.241\,r) \sqrt{1 - r}$$

2 Air flowing through an orifice experiences a definite transition point at or near the critical-pressure ratio. This can be seen from a graph of flow versus pressure ratio when plotted on elliptical coordinates. Here it can be seen that the rate of increase in flow for a decrease in back pressure drops off considerably in the critical region (see Fig. 5). This transition is also demonstrated by the characteristic of the flow coefficient, K_c. At or near the critical pressure ratio the function of K_c versus pressure ratio is seen to have a point of inflection (see Fig. 7). Finally, it is brought out by the characteristic of the expansion factor. The expansion factor is obviously an entirely different function in the critical region from what it is in the subcritical region (see Fig. 6).

3 The orifice flow characteristic for a zero approach velocity is an ellipse in the subcritical region.

4 The results of the author agree within 2 per cent of those of H. B. Reynolds; however, the shapes of the respective orifice characteristics vary. Reynolds expresses the entire characteristic with one equation. Stanton and Schiller indicate lesser flow rates in the critical region; however, the agreement is seen to be within $1^1/_2$ per cent. There is almost an exact agreement with the results of Bean, Buckingham, and Murphy for a range of r = 0.9 to r = 0.7, and a maximum variation of $1^1/_2$ per cent at r = 0.5. The subcritical flow equation agrees within 2 per cent of the data supplied by the ASME for 2-in. pipe at β = 0.1, and within 1 per cent for larger sizes of pipe.

TABLE 4 FLOW FACTORS AND FLOW COEFFICIENTS FOR SHARP-EDGED ORIFICES

Pressure ratio	Flow factor for air, Ω	Flow coefficient, K_c
1.0	0.000	...
0.98	0.093[a]	0.602[a]
0.96	0.130[a]	0.606[a]
0.94	0.159[a]	0.610[a]
0.92	0.182[a]	0.614[a]
0.90	0.203[a]	0.617[a]
0.85	0.246	0.631
0.80	0.279	0.641
0.75	0.308	0.655
0.70	0.332	0.670
0.65	0.354	0.687
0.60	0.373	0.710
0.55	0.389	0.731
0.53	0.395	0.742
0.50	0.403	0.756
0.45	0.414	0.776
0.40	0.423	0.795
0.35	0.430	0.809
0.30	0.436	0.820
0.25	0.441	0.828
0.20	0.444	0.832
0.15	0.446	0.839
0.10	0.448	0.840
0.05	0.449	0.843
0.00	0.449	0.843

[a] These are not experimental values and are shown for theoretical purposes only.

ACKNOWLEDGMENT

The author would like to acknowledge the valuable help and advice he received from Dr. R. A. Budenholzer.

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Discussion

STANFORD NEAL.[6] A need exists for more informiation concerning the performance of square-edged thin-plate orifices in the compressible and supersonic regions. The author presents data which confirm the general level of flow coefficient for orifices of low diameter ratio as given in the ASME Fluid Meters Report (1937).

As a by-product of another investigation by the writer's company, flow coefficients of several orifices were determined over a pressure-ratio range of 1.07 to 2.25 and are shown in the Fig. 9 herewith. Since the range of diameter ratios was from 0.20 to 0.90, the velocity of approach was an important factor as was not the case in the author's investigation. The flow coefficients agree reasonably well with the ASME recommended values for a diameter ratio of 0.20, but display increasingly large discrepancies as the ratio increases at low values of pressure ratio.

The highest value of diameter ratio given in the ASME report was 0.825, but the measured discrepancies are much larger than would be supposed from an error in extrapolation, and also exist for ratios of 0.65 and 0.80. The recommended values were calculated using the formula for Y up to pressure ratios of 2.25. The ASME report does not indicate the range over which the formula holds, but the agreement is better at pressure ratios of 2.00 than at 1.10.

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[6] Steam Research Section, Steam Turbine Engineering Division, General Electric Company, Schenectady, N. Y. Mem. ASME.

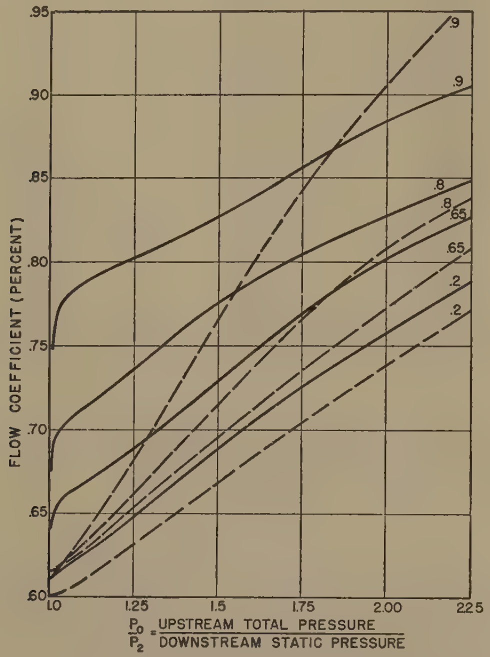

Fig. 9 Flow Coefficients for Sharp-Edged Flat-Plate Orifices in a 5-In-Diam Pipe With Vena-Contracta Pressure Taps

(Tests were made with air discharged to atmosphere. Flow coefficient is measured flow divided by the theoretical flow calculated from the total upstream and static downstream pressures. Solid lines from test data. Dashed lines calculated from ASME Fluid Meters Report.)

The flow coefficients were determined with vena-contracta taps which were placed in the recommended position. The length of downstream pipe was only 2 diam but it was observed that the stream filled the pipe discharge, except when the 0.20 orifice was used.

The tests were made carefully and it is believed that the accuracy is of the order of ±0.25 per cent. The data are shown here with the hope that they will be useful and may stimulate further investigation in this field.

An excellent subject for further graduate study would be the theoretical complement of the author's work. Techniques are now available for the study of the compressible subsonic and also the supersonic regions. It should be possible to flux-plot the subsonic and supersonic streams at different pressure ratios for both the two- and three-dimensional case. The change of vena-contracta size could be determined in the subsonic case, and the shock patterns could be determined in the supersonic case.

It is possible that the flow coefficient should approach 1.00 for an orifice of infinitesimal thickness at pressure ratios approaching infinity, as compared to the author's value of 0.84. It should be possible to determine the effect of plate thickness on flow coefficient by the same methods.

A further extension of the theoretical method should take into account the boundary layer and its effect on flow coefficient. If this problem is attacked from both the theoretical and practical aspects, the two methods should greatly extend our present knowledge.

Author's Closure

The next step to take in the study of critical flow would seem to be flow with velocity of approach. Mr. Neal's research shows coefficients down to pressure ratios as low as 0.444. These data should prove very valuable in this regard. Undoubtedly the problem would result in a much more complex solution with such variables as tap position, roughness of pipe, concentricity of the orifice, Reynolds number, etc., having a decided effect.

Air Flow Through Small Orifices in the Viscous Region[1]

By H. R. Linden[2] and D. F. Othmer[3]

Experimental data for air flow through square-edged circular orifices and their correlation with the pressure difference, the orifice diameter, and the air temperature are presented in nomographic form. The correlation holds well for small orifices with diameters ranging from $^1/_{16}$ to $^3/_{16}$ in., for pressure differences from 0.015 to 0.500 in. of water, and negligible velocity of approach. The air-temperature range within which the correlation can be applied is from 70 to 300 F, and corrections for barometric pressures from 27 to 32 in. of mercury can be made. The main use of this correlation will be found in its application to oil-burner design. By inclusion of the carbon-hydrogen-ratio criterion, the nomographic representation may be integrated with the system of combustion calculations, based upon the carbon-hydrogen ratio developed by the authors (1).[4] The nomograph enables the designer to calculate the theoretical fuel rate at which any hydrocarbon fuel can be burned per square inch of air-orifice area at a given pressure difference across orifices of known diameter and at specified air temperature, air pressure, and air humidity.

Nomenclature

The following nomenclature is used in the paper:

H = absolute humidity of air at orifice, lb of water vapor per lb of dry air
p = absolute pressure at orifice, in. of mercury (in. of water for experimental data)
p' = absolute pressure in wet-gas meter, in. of water
μ = absolute viscosity, centipoises
F_a = actual fuel rate, lb per hr
C = atmospheric correction factor
ρ_a = air density at orifice, pcf
c_1 = air-flow correction factor $= p'V/pV' = Q/Q'$
t = air temperature at orifice, deg F
t' = air temperature in wet-gas meter, deg F
R = carbon-hydrogen ratio of fuel
D = channel-of-approach diameter, in.
K = constant
Q = flow rate through orifice, cu ft per hr
Q' = flow rate through wet-gas meter, cu ft per hr
X = fraction excess air
V = humid volume at atmospheric pressure, cu ft per lb of dry air
A = orifice area, sq in.
C_o = orifice coefficient
d = orifice diameter, in.
ΔH = pressure difference across orifice (net draft), in. of water
M_o = rate of air flow, mols of air per sq in. orifice area per hr
r = ratio of plate thickness to orifice diameter
R_e = Reynolds number
F = theoretical fuel rate, lb per hr per sq in. orifice area
ρ_w = water density at micromanometer temperature, pcf

Introduction

This investigation of air flow through small orifices in the viscous region was initiated by design work on vaporizing pot-type oil burners (2). In such burners, fuel oil is vaporized in a pot by the heat of the flame above it while air is introduced into the pot through small orifices in its walls. The driving force for the flow of air is provided by either natural draft or a combination of natural draft and forced draft. The latter is usually produced by feeding air from a blower into a jacket around the pot. It was observed that the carbon-dioxide analyses of the flue gas were consistently lower than those calculated from the data of Tuve and Sprenkle (3), extrapolated to ratios of orifice diameter to channel of approach diameter d/D, below 0.2, although leakage of air had been eliminated and precise draft, orifice size, fuel rate, and fuel-composition data had been obtained. The effect of preheating as the air approaches the orifices was neglected and would have resulted in even higher calculated carbon-dioxide analyses if it had been taken into consideration.

An experimental investigation of air flow in the conventional flanged orifice-tube apparatus, fitted with radius taps, was undertaken for conditions approximating those encountered in vaporizing pot-type burner design, namely, air flow at practically atmospheric pressure through circular square-edged orifices from $^1/_{16}$ to $^1/_4$ in. diam, and pressure differences across the orifice, ranging from 0.01 to 0.50 in. of water at negligible velocities of approach. The results of this investigation, however, seemed of sufficient general interest to be presented as a study of viscous air flow in addition to a presentation suitable only for burner design. It was also realized that application to design calculations for natural- and liquefied-petroleum gas burners and mechanical-atomizing burners, introducing secondary air under similar flow conditions, may be possible.

A literature survey indicated that no reliable and consistent data were available for this region of fluid flow which is characterized by Reynolds numbers R_e, from 100 to 3000, and a ratio of orifice to channel of approach diameter d/D, below 0.2. The most generally accepted data for the viscous region of fluid flow are those by Tuve and Sprenkle (3). They are represented as a family of curves with the orifice coefficient plotted against R_e, at constant d/D. The orifice coefficients C_o, decrease at decreasing rate with decreasing values of d/D, and approach an apparent minimum near $d/D = 0.2$. From more recent work by Hira Lal Roy and N. K. Sen-Gupta (4) on free discharge from small orifices, it appeared that important changes in the characteristics of the semilog plot of orifice coefficient versus Reynolds number occur for very small orifices as the velocity of approach approaches zero. Although all previous investigators were able

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[1] This investigation is based upon work conducted at the Polytechnic Institute of Brooklyn, Brooklyn, N. Y.

[2] Supervisor, Oil Research Laboratory, Institute of Gas Technology, Technology Center, Chicago, Ill.

[3] Professor of Chemical Engineering, and Head of Department, Polytechnic Institute of Brooklyn, Brooklyn, N. Y. Mem. ASME.

[4] Numbers in parentheses refer to the Bibliography at the end of the paper.

Contributed by the Research Committee on Fluid Meters, and presented at the Annual Meeting, New York, N. Y., November 28–December 3, 1948, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society. Paper No. 48—A-93.

to correlate C_o with R_e by using d/D as a parameter, the orifice diameter d itself was used in this work. This approach seems justified by the data presented here which indicate the possibility that for small orifices (below $^1/_4$ in. diam), the parameter d/D may not hold generally. For instance, C_o increases as d/D approaches zero in contrast to existing data (3, 5, 6, 7, 8), but insufficient work has been done to determine if this is due to a reversal of trend below $d/D = 0.2$ or merely the effect of small orifice diameters.

Although the irregularities for these flow conditions would not be encountered frequently in industrial use of orifices for flow measurement, the effect on burner design may be considerable. Since the combustion of hydrocarbon fuels is very sensitive to a careful adjustment of excess air and air distribution, air-flow calculations on which design of such units is based should be quite precise. Design conditions usually specify fuel rate, chemical composition of fuel, maximum allowable excess air, and minimum available pressure differences across air-supply orifices. An empirical correlation based on the new experimental data of air flow through circular square-edged orifices, with orifice diameter, available pressure difference, and temperature, is developed here and combined with the C/H ratio for application to burner design. The final correlation is presented in nomographic form for direct use. Furthermore, the experimental data are also presented in the conventional plot of C_o versus R_e, for application to other gases in this range of Reynolds number.

EXPERIMENTAL WORK

The air-flow data were obtained with a 2-in-ID flanged orifice tube, provided with radius taps and with four orifice plates. The equipment was constructed from accurately machined stainless steel; and the orifices were measured with a microscope equipped with various calibrated eyepiece micrometers. The orifices appeared to be perfect under considerable magnification, and their dimensions are given in Table 1.

TABLE 1 DIMENSIONS OF ORIFICES

Orifice	Nominal size, in.	Diam. in.	Thickness of plate, in.	d/D
A.......	$^3/_{16}$	0.1888 ± 0.0004	0.037	0.094
B.......	$^1/_8$	0.1242 ± 0.0004	0.037	0.062
C.......	$^1/_{16}$	0.0636 ± 0.0004	0.037	0.032
D[a]......	$^1/_{16}$	0.0630 ± 0.0004	0.024	0.032

[a] Orifice D is not used in the correlation.

Flow rates were measured with a wet-gas meter; and constant air flow at all desired rates was obtained by reducing the pulsating suction from a small laboratory vacuum pump through a series of large vessels, orifices, needle valves, and bleed valves. The pressure difference across the radius taps was measured with a specially constructed, inclined glass-tube micromanometer employing a micrometer screw and sliding block of a 1-in. range and calibrated to 0.0005 in. In the low range, all readings were checked with a Bristol recording draft gage (oil-float type), which is calibrated to 0.002 and readable to 0.0005 in. of water. In addition, a simple inclined draft gage was used to make approximate adjustments before taking data. The micromanometer was especially designed and built for this investigation and used water as gage liquid. A schematic diagram of the experimental equipment and a drawing of the micromanometer are shown in Figs. 1 and 2.

The experimental data were taken to permit the calculation of the rate of air flow at the temperature, pressure, and humidity of the air entering the orifice, from the rate of flow, measured by the wet-gas meter at its temperature, its slightly lower pressure, and at complete saturation with water vapor. The data are summarized in Tables 2 to 5, inclusive. The orifice coefficients were calculated by the method used for incompressible fluids at negligible velocity of approach. Such a procedure was permissible, since, for this application, the ratio of pressure drop across the orifice to total pressure does not exceed 0.0015, and the ratio of orifice diameter to channel-of-approach diameter can be as high as 0.2 without affecting the assumption of negligible velocity of approach. From the flow rate through the orifice Q in cu ft per hr, the pressure difference across the orifice ΔH, in in. of water at a water density of ρ_w (in lb per cu ft), the air density ρ_a, at the orifice in lb per cu ft, and the diameter d, of the orifice in in., the orifice coefficient C_o can be derived as

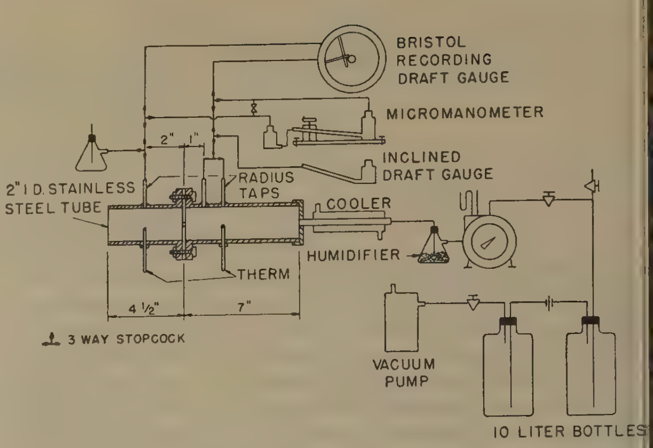


Fig. 1 Experimental Setup for Investigating Air Flow in Viscous Region

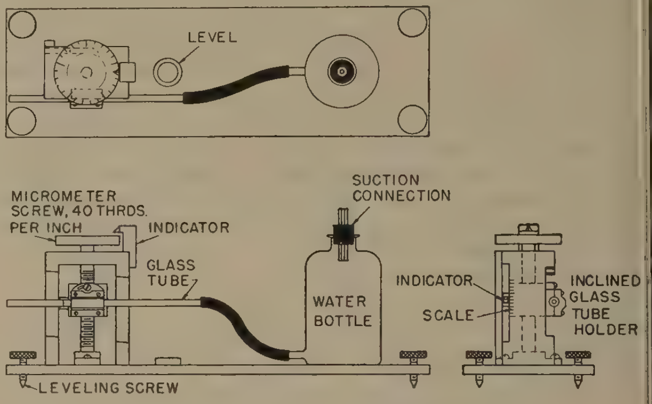


Fig. 2 Micromanometer

$$C_o = \frac{0.0220\, Q\, \rho_a^{0.5}}{d^2 \Delta H^{0.5}\, \rho_w^{0.5}} \qquad [1]$$

Reynolds number can be expressed as

$$\mathsf{R}_e = \frac{6.315\, Q\, \rho_a}{d\mu} \qquad [2]$$

where μ is the air viscosity at orifice conditions in centipoises. An average curve drawn through the best available viscosity-temperature data for air at atmospheric pressure was used. The maximum predicted error of C_o (as calculated from the maximum accidental errors of the various basic measurements) ranges from 4 per cent for $d = {}^1/_{16}$ and $\Delta H = 0.01$, to 0.8 per cent for $d = {}^3/_{16}$ and $\Delta H = 0.50$. The maximum deviation of the composite plot of C_o versus R_e, Fig. 3, from the most probable values is estimated to be within 2 per cent. When comparing these new data with the results obtained by previous investigators (3), the factors on page 769 should be noted:

TABLE 2 EXPERIMENTAL AND CALCULATED DATA FOR ORIFICE A

Flow Rate Wet Gas Meter Cu. ft./hr.	Absolute Humidity Entering Air	Temp. of Air at Orifice; °F	Corr.Barom Pressure; Inches of Water	Pressure Drop Across Orifice Inches of Water	Temp. of Wet Gas Meter; °F	Pressure in Wet Gas Meter Inches of Water	Air Flow Correction	Wet Gas Meter Calibration	Corr.Flow Rate; Cu. ft./hr.	Air Density at Orifice; lbs/cu.ft.	Air Viscosity at Orifice; centipoises	Reynolds Number	Orifice Coefficient
Q'	H_1	t_1	p_1	ΔH	t_2	p_2	c_1	c_2	Q	ρ_a	μ	Re	C_o
3.185	.0059	78.0	406	.0092	77.5	406	.980	.990	3.09	.0735	.0183	415	.684
3.325	.0095	81.0	409	.0100	79.0	409	.984	.990	3.24	.0734	.0184	458	.689
3.34	.0098	81.0	409	.0100	79.0	409	.984	.990	3.25	.0734	.0184	460	.691
3.35	.0059	78.5	406	.0100	77.5	406	.980	.990	3.25	.0734	.0184	434	.692
3.445	.0060	80.5	407	.0100	80.0	407	.976	.990	3.33	.0733	.0184	445	.706
3.545	.0030	75.5	412	.0105	77.0	412	.972	.990	3.41	.0751	.0183	468	.713
4.125	.0097	81.5	409	.0150	79.5	409	.984	.990	4.01	.0733	.0185	532	.694
4.18	.0098	82.5	409	.0150	80.5	409	.982	.990	4.06	.0731	.0185	537	.703
4.23	.0098	82.0	409	.0150	80.0	409	.984	.990	4.11	.0732	.0185	545	.711
4.30	.0060	81.0	407	.0150	80.5	407	.976	.990	4.16	.0732	.0184	554	.719
4.79	.0060	77.5	406	.0200	77.0	406	.980	.990	4.64	.0735	.0183	622	.696
4.79	.0099	83.0	409	.0200	80.5	409	.982	.990	4.65	.0731	.0185	615	.697
4.81	.0098	83.0	409	.0200	80.5	409	.982	.990	4.67	.0731	.0185	617	.700
5.84	.0047	78.6	409	.0300	78.5	409	.975	.990	5.63	.0740	.0184	758	.692
7.385	.0047	78.5	409	.0500	78.5	409	.975	.990	7.12	.0740	.0184	958	.678
8.62	.0047	78.5	409	.0700	78.3	409	.975	.990	8.32	.0740	.0184	1120	.670
10.08	.0058	80.0	407	.0970	80.0	406	.973	.990	9.71	.0733	.0184	1296	.661
10.19	.0050	78.0	409	.100	78.0	408	.973	.990	9.82	.0741	.0184	1330	.663
12.26	.0050	77.5	409	.150	78.0	408	.972	.990	11.79	.0741	.0183	1594	.650
14.02	.0050	77.2	409	.200	77.5	408	.975	.990	13.53	.0741	.0183	1829	.645
15.63	.0055	80.0	407	.250	80.0	406	.973	.990	15.05	.0734	.0184	2010	.640
16.93	.0055	78.5	406	.300	77.0	404	.977	.990	16.36	.0735	.0184	2185	.635
18.20	.0051	79.0	407	.350	79.0	405	.970	.990	17.46	.0735	.0184	2340	.626
19.36	.0048	78.8	412	.400	77.0	410	.977	.990	18.73	.0745	.0184	2530	.632
21.5	.0048	79.0	412	.500	77.0	410	.977	.990	20.8	.0745	.0184	2820	.629

TABLE 3 EXPERIMENTAL AND CALCULATED DATA FOR ORIFICE B

Flow Rate Wet Gas Meter Cu. ft./hr.	Absolute Humidity Entering Air	Temp. of Air at Orifice; °F	Corr.Barom Pressure; Inches of Water	Pressure Drop Across Orifice Inches of Water	Temp. of Wet Gas Meter; °F	Pressure in Wet Gas Meter Inches of Water	Air Flow Correction	Wet Gas Meter Calibration	Corr.Flow Rate; Cu. ft./hr.	Air Density at Orifice; lbs/cu.ft.	Air Viscosity at Orifice; centipoises	Reynolds Number	Orifice Coefficient
Q'	H_1	t_1	p_1	ΔH	t_2	p_2	c_1	c_2	Q	ρ_a	μ	Re	C_o
1.461	.0047	72	409	.0095	74.0	409	.976	.990	1.413	.0748	.0182	295	.720
1.469	.0130	79	410	.0093	80.5	410	.981	.990	1.425	.0737	.0184	290	.727
1.531	.0114	85	412	.0100	82.0	412	.983	.990	1.490	.0733	.0185	300	.731
1.549	.0121	85	411	.0100	85.0	411	.978	.990	1.498	.0731	.0185	300	.734
1.628	.0130	79	410	.0113	80.5	410	.981	.990	1.580	.0737	.0184	321	.731
2.182	.0121	85	411	.0197	85.0	411	.978	.990	2.11	.0731	.0185	423	.736
2.57	.0169	79	408	.0287	78.0	408	.994	.990	2.53	.0732	.0184	471	.734
2.65	.0121	85	411	.0299	85.0	411	.978	.990	2.57	.0731	.0185	514	.727
2.955	.0169	80	408	.0395	78.5	408	.995	.990	2.91	.0730	.0184	586	.719
3.385	.0121	85	411	.0493	85.0	411	.978	.990	3.28	.0731	.0185	656	.723
3.885	.0169	80	408	.0705	78.5	408	.995	.990	3.84	.0730	.0184	774	.708
4.01	.0121	85	411	.0710	85.0	411	.978	.990	3.88	.0731	.0185	778	.713
4.69	.0121	85	411	.1010	85.0	411	.978	.990	4.54	.0731	.0185	910	.700
4.97	.0170	80	408	.120	78.5	408	.995	.990	4.89	.0730	.0184	985	.691
5.02	.0089	84	406	.120	83.0	406	.977	.990	4.86	.0726	.0185	969	.686
5.56	.0111	87	409	.150	87.5	409	.972	.990	5.43	.0725	.0185	1080	.686
5.575	.0096	81	410	.150	82.0	410	.976	.990	5.38	.0735	.0184	1093	.683
6.415	.0111	87	409	.200	87.5	408	.970	.990	6.16	.0725	.0185	1226	.673
7.72	.0111	87	409	.300	87.5	408	.970	.990	7.41	.0725	.0185	1474	.662
8.68	.0094	87	412	.400	82.0	411	.985	.990	8.47	.0732	.0186	1592	.657
8.80	.0111	87	409	.400	87.0	408	.970	.990	8.45	.0725	.0185	1683	.653
9.28	.0109	80	402	.466	81.0	401	.977	.990	8.97	.0722	.0184	1788	.641
9.65	.0087	82	410	.500	82.5	409	.973	.990	9.28	.0734	.0185	1870	.644
9.77	.0087	81	410	.500	82.5	409	.972	.990	9.40	.0736	.0184	1910	.654
10.50	.0091	81	410	.600	82.0	409	.972	.990	10.10	.0735	.0184	2050	.641

TABLE 4 EXPERIMENTAL AND CALCULATED DATA FOR ORIFICE C

Flow Rate Wet Gas Meter Cu. ft./hr.	Absolute Humidity Entering Air	Temp. of Air at Orifice; °F	Corr.Barom Pressure; Inches of Water	Pressure Drop Across Orifice Inches of Water	Temp. of Wet Gas Meter; °F	Pressure in Wet Gas Meter Inches of Water	Air Flow Correction	Wet Gas Meter Calibration	Corr.Flow Rate; Cu. ft./hr.	Air Density at Orifice; lbs/cu.ft.	Air Viscosity at Orifice; centipoises	Reynolds Number	Orifice Coefficient
Q'	H_1	t_1	p_1	ΔH	t_2	p_2	c_1	c_2	Q	ρ_a	μ	Re	C_o
.3655	.0050	82.0	412	.0100	80.5	412	.979	.990	.354	.0739	.0185	140	.664
.374	.0048	81.5	412	.0100	79.5	412	.979	.930	.362	.0740	.0185	144	.679
.466	.0055	75.0	416	.0150	76.0	416	.975	.990	.450	.0757	.0183	185	.695
.5435	.0039	81.5	415	.0200	81.0	415	.970	.990	.521	.0745	.0184	209	.694
.555	.0050	81.8	412	.0200	80.0	412	.979	.990	.538	.0739	.0185	213	.713
.670	.0039	82.0	415	.0300	80.7	415	.972	.990	.645	.0745	.0185	262	.701
.785	.0030	73.3	416	.0400	73.5	416	.978	.990	.760	.0760	.0182	316	.721
.8865	.0040	82.5	415	.0500	81.0	415	.972	.990	.835	.0745	.0185	334	.718
1.077	.0041	82.2	415	.0700	80.6	415	.972	.990	1.035	.0745	.0185	414	.735
1.286	.0041	82.0	415	.100	80.5	415	.972	.990	1.236	.0745	.0185	494	.735
1.592	.0041	80.8	415	.150	80.0	415	.970	.990	1.530	.0746	.0184	617	.745
1.803	.0048	79.5	415	.196	80.5	415	.973	.990	1.735	.0748	.0184	701	.740
2.246	.0055	78.0	415	.300	80.0	415	.972	.990	2.16	.0750	.0184	875	.744
2.53	.0030	73.5	416	.400	74.0	416	.978	.990	2.45	.0760	.0182	1014	.736
2.805	.0030	73.3	416	.500	73.5	415	.975	.990	2.705	.0760	.0182	1120	.725
3.045	.0030	73.0	416	.600	73.5	415	.975	.990	2.94	.0760	.0182	1220	.720
3.38	.0055	74.5	416	.750	75.5	415	.973	.990	3.255	.0757	.0183	1336	.711

CORRECTIONS TO TABLES 3 AND 5 OF THIS PAPER

In Table 3, first column, under Q', the tenth line from the bottom should read 5.65. Also in Table 3, thirteenth column under $\mathbf{R}_e$, the sixth line from the bottom should read 1692. In Table 5, fifth column under ΔH, the tenth line from the bottom should read .0200.

TABLE 5 EXPERIMENTAL AND CALCULATED DATA FOR ORIFICE D

Flow Rate Wet Gas Meter Cu. ft./hr.	Absolute Humidity Entering Air	Temp. of Air at Orifice; °F	Corr.Barom Pressure; Inches of Water	Pressure Drop Across Orifice Inches of Water	Temp. of Wet Gas Meter; °F	Pressure in Wet Gas Meter Inches of Water	Air Flow Correction	Wet Gas Meter Calibration	Corr.Flow Rate; Cu. ft./hr.	Air Density at Orifice; lbs/cu.ft.	Air Viscosity at Orifice; centipoises	Reynolds Number	Orifice Coefficient
Q'	H_1	t_1	p_1	ΔH	t_2	p_2	c_1	c_2	Q	ρ_a	μ	Re	C_o
.362	.0050	77.8	412	.0100	78.0	412	.975	.990	.349	.0746	.0184	142	.669
.3715	.0031	77.9	411	.0100	76.2	411	.978	.990	.359	.0745	.0183	147	.688
.3755	.0031	77.5	411	.0100	76.2	411	.978	.990	.363	.0745	.0183	148	.696
.548	.0048	74.5	414	.0100	75.0	414	.977	.990	.530	.0755	.0183	219	.722
.5535	.0031	77.5	411	.0200	76.2	411	.978	.990	.535	.0745	.0183	218	.725
.793	.0050	78.0	412	.0400	78.0	412	.975	.990	.765	.0746	.0184	311	.732
1.056	.0046	74.5	414	.0700	75.0	414	.977	.990	1.020	.0755	.0183	422	.743
1.130	.0031	77.5	411	.0800	76.2	411	.978	.990	1.093	.0745	.0183	447	.741
1.422	.0050	78.5	412	.125	78.0	412	.976	.990	1.359	.0745	.0184	552	.736
1.752	.0045	74.5	414	.200	75.0	414	.977	.990	1.694	.0755	.0183	702	.730
2.42	.0050	78.8	412	.400	78.0	412	.978	.990	2.34	.0745	.0184	951	.709
2.66	.0045	74.5	414	.500	75.0	414	.977	.990	2.58	.0755	.0183	1065	.702
3.19	.0050	79.0	412	.750	78.0	411	.978	.990	3.09	.0745	.0184	1255	.684

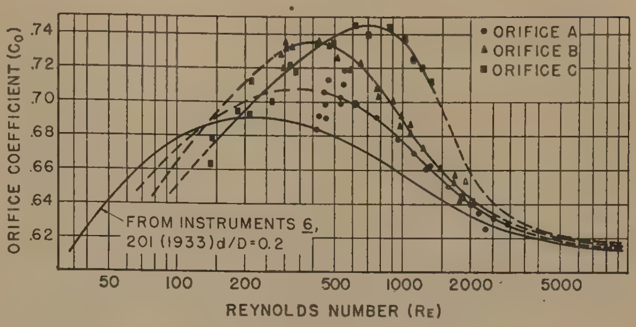


FIG. 3 COMPOSITE PLOT OF ORIFICE COEFFICIENT VERSUS REYNOLDS NUMBER

1 Due to a negligible velocity of approach (d/D below 0.1) identical data would have been obtained with corner or vena-contracta taps and possibly even with pipe taps.

2 The ratios of thickness of orifice plate to orifice diameter r, are larger than used or recommended by any previous investigator (3, 9, 10, 11), and it was found that the shift to the right of the C_o versus R_e plots with decreasing diameters, i.e., increasing r, may be due to these large values of r, especially for orifice C. A plot of C_o versus R_e for an orifice D, of approximately equal diameter as C, but in a thinner plate, indicated a shift of the optimum (C_o = 0.744) from R_e = 700 to R_e = 400 for a corresponding decrease of r from 0.59 to 0.39.

The equation correlating the experimental data for the average conditions of 80 F and a humidity of 0.007 lb of water vapor per lb of dry air at 410 in. of water absolute pressure is

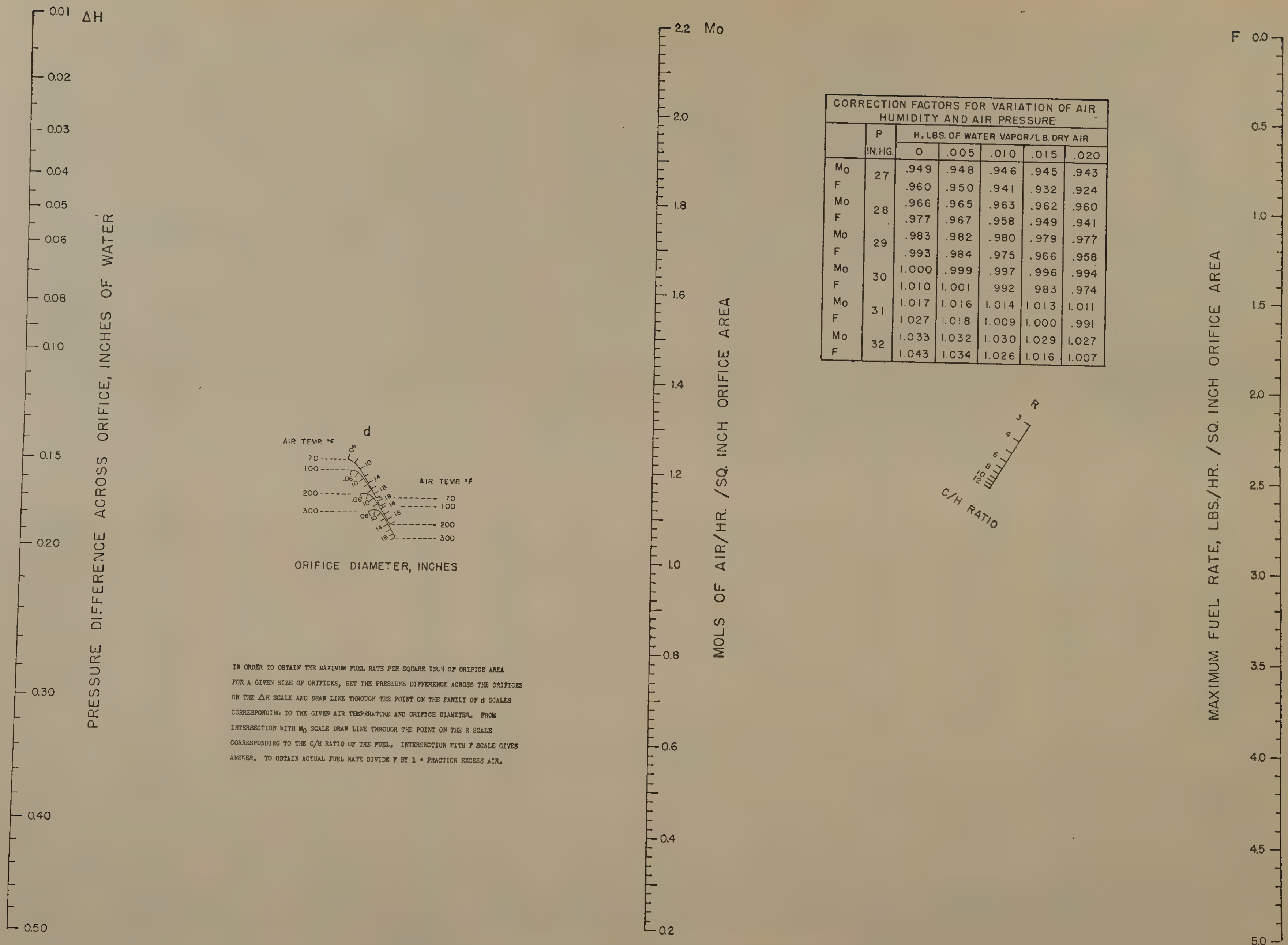
$$(Q)\ 80\ \text{F} = 609\ d^{1.84}\ \Delta H^{0.4685 - \frac{0.0051}{d}} \ldots\ldots [3]$$

The maximum deviation of this equation from the average experimental data is ±4 per cent. The development of the expressions used as the basis for the nomograph, Fig. 4, and the basis for the humidity and barometric pressure correction are given in the Appendix.

USE OF CORRELATION

The air-flow correlation probably is most useful in the design of vaporizing pot-type burners. For this type of oil burner, the combustion air is introduced into the oil-vaporizing and combustion zone (the pot) through small orifices by the driving force of either natural draft or forced draft; the latter supplied by a fan through a jacket around the pot. Large numbers of such units are produced for space heaters and central heating systems (12), and considerable activity in this field has been apparent because of changes in the distillate fuel-oil supply (13). The nomograph, Fig. 4, should be of considerable help since all such design problems can be simplified by its use. In order to calculate the maximum fuel rate for a forced-draft unit, the air pressure in the jacket is added to the draft over the fire (maintained relatively constant by a draft regulator) and set on the ΔH scale. The average air temperature is then estimated from previous experimental data, and the proper d scale selected.

If more than one-size orifice is used, the various diameters should be averaged by weighting each diameter with the fraction of the total orifice area contributed by the orifices of the corresponding size. If elliptical orifices are used, the square root of the product of the two axes should be substituted for the diameter. By drawing a straight line from ΔH through d, the intersection with the M_o scale gives the mols of air per hr per sq in. of orifice area. A straight line from M_o through the C/H ratio of the fuel (1, 13, 14), set on the R scale gives the corresponding



CORRECTION FACTORS FOR VARIATION OF AIR HUMIDITY AND AIR PRESSURE

	P	H, LBS. OF WATER VAPOR/LB. DRY AIR				
	IN. HG.	0	.005	.010	.015	.020
M_0	27	.949	.948	.946	.945	.943
F		.960	.950	.941	.932	.924
M_0	28	.966	.965	.963	.962	.960
F		.977	.967	.958	.949	.941
M_0	29	.983	.982	.980	.979	.977
F		.993	.984	.975	.966	.958
M_0	30	1.000	.999	.997	.996	.994
F		1.010	1.001	.992	.983	.974
M_0	31	1.017	1.016	1.014	1.013	1.011
F		1.027	1.018	1.009	1.000	.991
M_0	32	1.033	1.032	1.030	1.029	1.027
F		1.043	1.034	1.026	1.016	1.007

IN ORDER TO OBTAIN THE MAXIMUM FUEL RATE PER SQUARE INCH OF ORIFICE AREA FOR A GIVEN SIZE OF ORIFICES, SET THE PRESSURE DIFFERENCE ACROSS THE ORIFICES ON THE ΔH SCALE AND DRAW LINE THROUGH THE POINT ON THE FAMILY OF d SCALES CORRESPONDING TO THE GIVEN AIR TEMPERATURE AND ORIFICE DIAMETER. FROM INTERSECTION WITH M_0 SCALE DRAW LINE THROUGH THE POINT ON THE R SCALE CORRESPONDING TO THE C/H RATIO OF THE FUEL. INTERSECTION WITH F SCALE GIVES ANSWER. TO OBTAIN ACTUAL FUEL RATE DIVIDE F BY 1 • FRACTION EXCESS AIR.

FIG. 4 NOMOGRAPH FOR DETERMINING AIR FLOW FOR OIL-BURNER DESIGN

theoretical fuel rate. Divided by 1 plus the fraction excess air considered the minimum for trouble-free combustion and correcting for local atmospheric conditions (elevation, etc.) gives the maximum fuel rate per square inch of orifice area. The maximum fuel rate per hour for the given unit is then obtained by multiplying by the total orifice area.

In this manner any of the variables can be evaluated, and such a burner designed specifically for a given fuel and fuel rate, if the limits of available pressure drop (net draft) and excess-air requirements are known.

Design problems are simplified further by the use of a nomographic system of combustion calculations previously presented (1), and based upon the carbon-hydrogen ratio of the fuel. The C/H ratio can be obtained empirically from the Diesel Index, and the 50 per cent ASTM boiling point for most liquid-petroleum fuels (1, 13, 14).

For the design of natural-draft units, correlations between stack temperature, stack diameter, and stack draft should also be considered (15). The relationship of stack draft and draft over the fire depends upon the design of the combustion chamber and the heat exchanger, but usually can be estimated from the performance of existing units. A schematic drawing of a vaporizing pot-type burner is shown in Fig. 5.

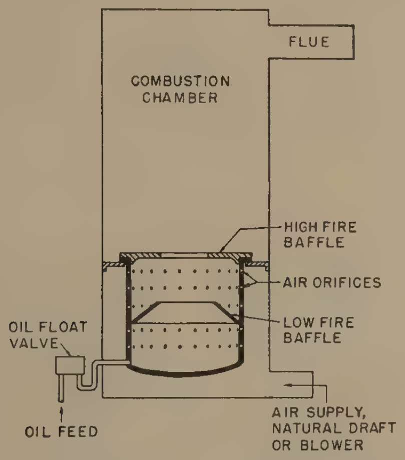


Fig. 5 Schematic Diagram of Vaporizing Pot-Type Burner

For the calculation of fan-discharge-pressure requirements the nomograph can be used in reverse. The theoretical fuel rate to be set on scale F in Fig. 4 can be calculated from the total orifice area A, the atmospheric correction factor from Table 6, C, the fraction excess air X, and the actual fuel rate F_a, expressed in lb per hr

$$F = \frac{F_a(1 + X)}{AC} \dots\dots\dots\dots\dots\dots [4]$$

For application of the orifice-coefficient data to other gases, the composite plot of C_o versus R_e, Fig. 3, should be used in conjunction with Equations [1] and [2]. It should be noted that the parameter d/D may not hold for the orifices investigated here and that application should be restricted to orifices between $^1/_{16}$ and $^3/_{16}$ in. diam and d/D below 0.2, in the range of Reynolds number shown in solid lines in Fig. 3.

Acknowledgment

Thanks are due to W. M. Holaday, T. G. Roehner, E. B. Delgass, and B. S. Garrett, of the Socony-Vacuum Laboratories, for their help in making the necessary equipment and laboratory space available, and for their comments, suggestions, and varied technical assistance.

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Appendix

The high-temperature data were calculated from the experimental room-temperature data by using the perfect-gas laws, the plot of variation of air viscosity with temperature based on most available recent data, and Equation [3]. The empirical relationship of the four variables was determined by standard graphical methods (16), and patterned after the theoretical equation, i.e., the equation neglecting the effects of viscosity

$$Q = K\,(460 + t)^{1/2}\,\Delta H^{1/2} d^2 \dots\dots\dots\dots\dots [5]$$

A composite log-log plot of Q versus ΔH at 80, 200, and 300 F for the orifices was used as the basis for developing the required equations. The actual equation applicable to the flow of air through square-edged, circular orifices at pressure differences from 0.01 to 0.50 in. of water, orifice diameters from 0.06 to 0.20 in., and temperatures from 70 to 300 F, is

$$Q = 30.43\,(320 + t)^{0.5}\,\Delta H^{0.4685}\,d^{1.84} - \frac{0.00036\,t^{0.606}}{d} \dots [6]$$

The maximum deviation of Equation [6] from the average experimental data is approximately 4 per cent. For general use, the most practical form of Equation [6] expresses the air flow in mols of air per hr per sq in. of orifice area (M_o)

$$M_o = \frac{53.6\ (t + 320)^{0.5}\ \Delta H^{0.4685}}{(t + 460)\ d^{0.16}} - \frac{0.000634\ t^{0.606}}{(t + 460)\ d^3} \ldots\ldots [7]$$

Equation [7] is presented in nomographic form in Fig. 4. A family of equations at various constant temperatures is shown in the form

$$(M_o)_t = f_1(d)\, f(\Delta H) + f_2(d) \ldots\ldots\ldots\ldots [8]$$

Also included in the nomograph is a function expressing the theoretical fuel rate F, of hydrocarbon fuels in terms of the air flow per square inch of orifice area M_o, and the C/H ratio of the fuel R

$$F = (0.989)\ (0.209\ M_o)\ \frac{12\ (R + 1)}{R + 3} \ldots\ldots\ldots [9]$$

The factor 0.989 corrects for the humidity of the air. Equations [6] to [9] are based upon an atmospheric pressure p, of 410 in. of water (30.15 in. of mercury), and a humidity H, of 0.007 lb of water per lb of dry air. This corresponds to the average atmospheric conditions for the experimental data. For other atmospheric conditions, corrections can be made on the assumption that normal atmospheric variations of air density at constant air temperature will not affect the orifice coefficient. The airflow and fuel-rate corrections may then be expressed by the following equations

$$\text{Correction factor for } M_o = \sqrt{\frac{p(1-0.612H)}{30.0}} \ldots\ldots\ldots\ldots [1$$

$$\text{Correction factor } (C) \text{ for } F = \frac{\sqrt{p(1 - 0.612\ H)}}{5.415\ (1 + 1.61\ H)} \ldots\ldots\ldots [11$$

Correction factors for the nomograph in Fig. 4 for variation of air humidity and atmospheric pressure are given in Table 6

TABLE 6 CORRECTION FACTORS FOR VARIATION OF AI HUMIDITY AND AIR PRESSURE

	p in. Hg	H, lb of water vapor per lb of dry air				
		0	0.005	0.010	0.015	0.02
M_o	27	0.949	0.948	0.946	0.945	0.94
F		0.960	0.950	0.941	0.932	0.92
M_o	28	0.966	0.965	0.963	0.962	0.96
F		0.977	0.967	0.958	0.949	0.94
M_o	29	0.983	0.982	0.980	0.979	0.97
F		0.993	0.984	0.975	0.966	0.95
M_o	30	1.000	0.999	0.997	0.996	0.99
F		1.010	1.001	0.992	0.983	0.97
M_o	31	1.017	1.016	1.014	1.013	1.01
F		1.027	1.018	1.009	1.000	0.99
M_o	32	1.033	1.032	1.030	1.029	1.02
F		1.043	1.034	1.026	1.016	1.00

Reaction Tests of Turbine Nozzles for Supersonic Velocities

By J. H. KEENAN,[1] CAMBRIDGE, MASS.

A machine for testing turbine nozzles by the reaction method, which was described in a previous paper,[2] was used to test a series of convergent-divergent turbine nozzles. The results of these tests, along with the test of a convergent turbine nozzle, are compared with each other and with analytical values. Two kinds of analytical values are employed, namely, the usual values obtained from an assumed isentropic expansion from inlet state to exhaust pressure, and the values obtained from the assumption that the processes in the nozzle are isentropic except for a normal shock which takes up a position in the nozzle such as to cause the stream to fill the exit area at the exhaust pressure whenever possible. This latter kind of analytical value involves no shock when the exit area can be filled at the exhaust pressure by means of isentropic processes only, or when the exhaust pressure is lowered so far that the shock has passed out of the passage. The agreement of the test results with the calculated results of this latter kind is good, and the disagreement which exists can be attributed largely to separation at the shock and to transmission of exhaust-pressure effects upstream through the boundary layer.

Introduction

In the year 1920 the Steam Turbine Department of the General Electric Company in Schenectady, N. Y., constructed a machine for testing turbine nozzles with steam by the reaction method. Over a period of several years of testing the machine was modified to eliminate inherent errors, and the resulting device was described and sample test results were given in a previous paper,[2] before this Society.

Although publication of further test results from this machine has been delayed many years by a deflection of the interests of those who developed the machine to other activities, it appears that no subsequently published material has made these data obsolete. Indeed, the evolution of the gas turbine in its various applications has made much of this material of more general value today than it was 20 years ago.

Description of Apparatus

The testing machine and its operation were described in detail in the previous publication[2] from which Figs. 1 and 2 are reproduced. Fig. 1 shows a side elevation of the machine in a plane parallel to the direction of flow from the nozzle. The nozzle group to be tested is bolted to the bottom flange of a vertical pipe in such a way as to cause the nozzles to discharge horizontally. This vertical pipe is hung from a hollow horizontal shaft with which it forms a vertical T. The shaft is supported at each end on ball bearings. The moment about the bearing axis of the reaction force at the nozzle is transmitted through a horizontal arm on the shaft to sensitive platform scales.

The exhaust-chamber shell extends up to and envelops the shaft. Steam is admitted at one end of the enveloping cylinder and immediately enters the shaft through radial holes. It flows down through the vertical pipe, through the nozzle, into the exhaust chamber, thence to a surface condenser. From the condenser the condensate line leads through a pump to weighing tanks.

Fig. 2 shows a model of the exhaust chamber, including a screen pad, the prototype of which was used in the full-scale machine to prevent dynamic action of exhaust steam on the nozzle arm. The pad consists of a series of coarse screen disks, with small central holes, set parallel to each other in that part of the chamber between the nozzle and the exhaust pipe. An annulus of fine screen is secured to each disk, concentric with it, and fitted closely to the chamber wall.

Validity of Test Results

In the previous paper,[2] the sources of error in measurement of reaction were discussed at some length. The principal problem was to attain a quiescent condition in the exhaust chamber so that extraneous forces would not be applied to the nozzle arm of the machine. Extensive evidence presented there indicates that this problem was solved adequately by means of the screen pad.

In the authors' closure to the discussion of that paper the principal sources of error in flow measurement were discussed at some length. The precautions taken included various tests for leakage into or out of the machine, and these indicated that the precautions were adequate.

The applicability of these test results to nozzles in a turbine assembly is a question entirely separate from that of the precision of measurement of the quantities involved in the test. The measurements yield the mass flow and the momentum flux from the nozzle group or cascade for certain essentially uniform inlet stagnation conditions and for exhaust into a space of uniform pressure. The velocity deduced from these two quantities is a hypothetical one. It is the velocity of a uniform stream having the mass flow and momentum flux of the actual stream. The kinetic energy of such a stream would be somewhat less than that of the actual stream because the actual stream is nonuniform in velocity. Whether the velocity to be employed in the velocity diagram of a turbine stage should be selected to accord with the momentum of the actual stream or with the kinetic energy is a question that need not be answered here. Suffice it to say that as a rule the nozzle which produces greater momentum per unit mass of fluid will give greater energy per unit mass as well.

The second consideration is that the nozzle group tested is arranged in an arc of infinite radius and of finite length The extent to which turbine nozzles depart from these conditions varies from turbine to turbine and from stage to stage within the same turbine. The number of nozzles in a test group varied from three to sixteen and was commonly four or five. Those with the larger numbers should give results more nearly approximat-

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[1] Professor of Mechanical Engineering, Massachusetts Institute of Technology. Mem. ASME.

[2] "A Machine for Testing Steam-Turbine Nozzles by the Reaction Method," by G. B. Warren and J. H. Keenan, Trans. ASME, vol. 48, 1926, pp. 33–64.

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society. Paper No. 48—A-61.

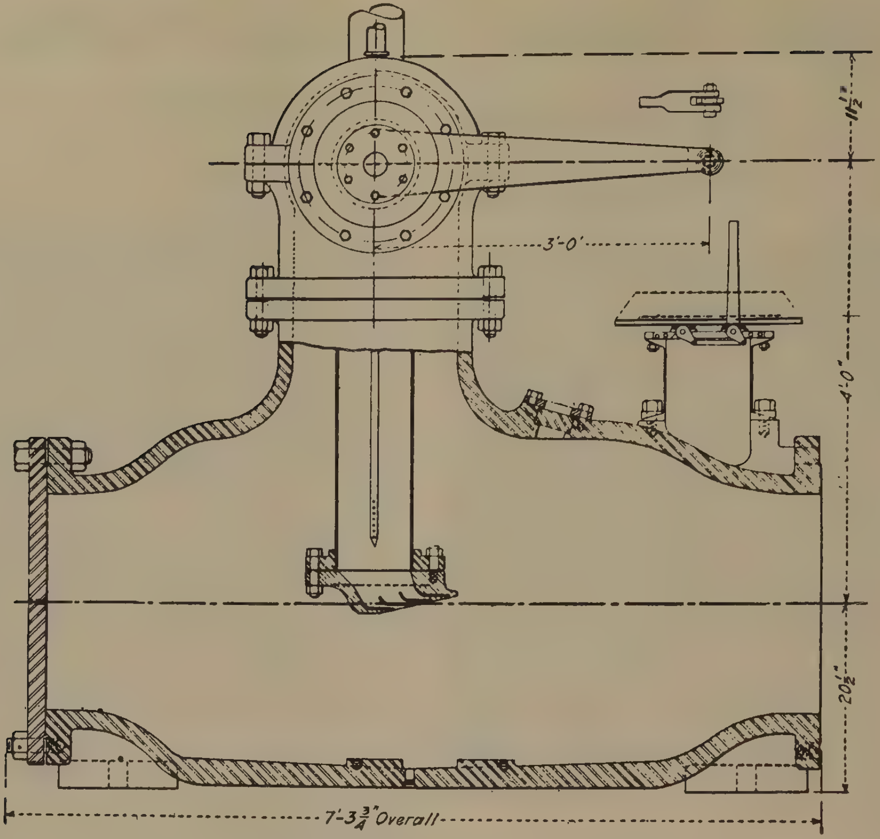

Fig. 1 Nozzle Testing Machine, Side Elevation

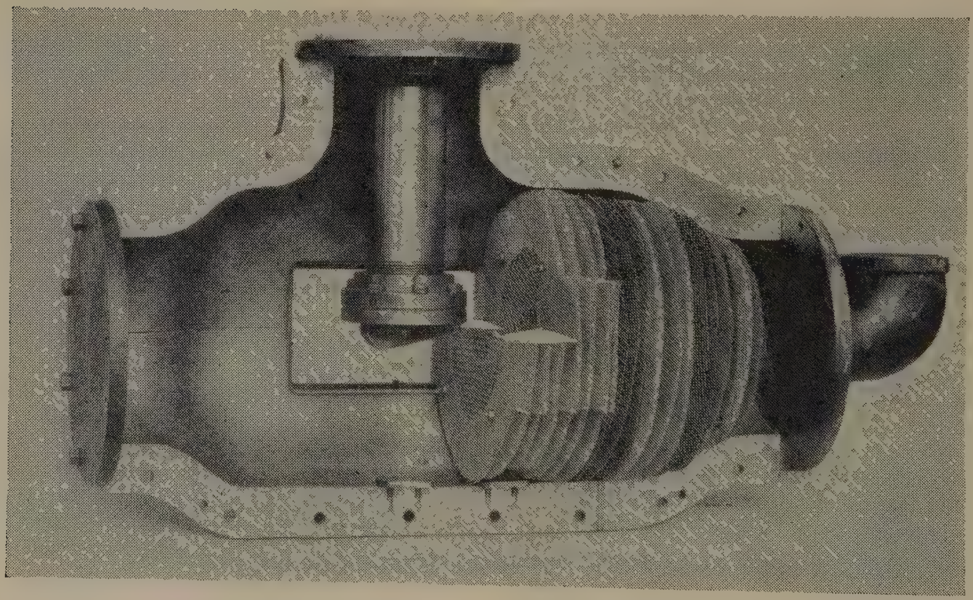

Fig. 2 Model of Exhaust Chamber With Screens Installed
(The prototype of the model of the screen pad was used to eliminate dynamic action of exhaust steam on the nozzle arm.)

ing the endless arc, although for supersonic exit conditions the approximation would probably be good in any event.

Although the importance of conditions adjacent to the nozzle-exit openings was not as fully understood at the time of these tests as now, a reasonable effort was made to minimize the effect of these conditions. Downstream from the exit openings, flat surfaces were generally avoided. Alongside the opening the surfaces were generally commensurate with those found in steam turbines.

Where the radial height of the nozzle was small it was sometimes necessary to have two parallel rows of nozzles in order to make the flow and reaction magnitudes large enough for precise measurement. The distance between rows was made 4 to 7 times the radial height, and the intervening surface was recessed to permit entrainment of fluid as freely as possible. (An exception to this rule is the reamed nozzle with a divergence ratio of 7.9 in which the two rows were immediately adjacent to each other as in the turbine for which the nozzle was designed.) It is possible, nevertheless, that some small difference would be found between test results for a single row and for a double row, because of the restriction of entrainment between rows, although such a comparison was never actually made. A comparison of characteristics of test results obtained for single rows and double rows reveals no consistent difference which could be attributed to the difference in number of rows.

Inlet turbulence and velocity distribution were not as carefully controlled as they would have been had our present knowledge of the effects of these things existed at the time of the tests. Reasonable precautions were taken, nevertheless. The cross-sectional area of the approach pipe was more than 10 times as great as the area of the largest nozzle and was more than 50 times as great as the areas of some of the nozzles.

In order to test whether a more carefully controlled inlet condition would modify the test results, a nozzle of large area was tested with and without a controlled inlet-velocity distribution and turbulence. Control was applied by inserting into the vertical approach pipe several groups of orifices in series, followed by baffles and screens which redistributed the flow over the entire pipe cross section. This device was tested with air for uniformity of distribution before it was inserted in the machine. Tests made under the controlled conditions gave velocity coefficients less than 0.25 per cent greater than those with the uncontrolled condition, a difference representing about the normal degree of reproducibility of the test results.

This smallness of the difference resulting from controlling the inlet-velocity distribution can doubtless be attributed to a relatively high degree of turbulence in the inlet pipe. The value of the Reynolds number, based on pipe diameter, covered a range from 0.9×10^5 to 6×10^5 in the vertical inlet pipe.

For certain nozzles, reproducibility of results was of a distinctly lower order than for the general run of nozzles. This characteristic may be attributable to an instability in the flow. Although the incidence of this apparent instability was irregular, it seemed to be confined to the mechanically weaker nozzle structures, that is, it never occurred in nozzles having sturdy partitions between passages, such as those in convergent-divergent nozzles. It seems probable that flexibility of the nozzle partitions may have permitted more than one partially stable flow pattern.

The nozzle was oriented in the machine so that the most probable direction of the issuing stream would be perpendicular to the axis of the vertical approach pipe. For converged nozzles with a flat surface on the nozzle partition downstream from the nozzle throat, this flat surface was placed perpendicular to the approach-pipe axis.

If the center of the issuing stream had always been vertically below the bearings, then the measured reaction would have been the horizontal component of the resultant reaction. Because of the turning of the stream in the nozzle, however, the center of the issuing stream was always slightly off the vertical line through the bearings in the direction of the issuing velocity.[3] This resulted in a measured reaction slightly greater than the horizontal component for upward deflection and slightly less for downward deflection of the jet. For upward deflection, as for a convergent nozzle at subsonic velocities, this effect caused the measured reaction to approximate the resultant reaction within 0.1 per cent in virtually all cases. For downward deflection, as for a convergent nozzle at supersonic velocities, this effect caused the measured reaction to be less than the horizontal component by perhaps 1 per cent at a Mach number of 1.5. It may be said, therefore, that the measured reaction is very nearly the resultant reaction for convergent nozzles at Mach numbers less than about 1.25 and for convergent-divergent nozzles for Mach numbers less than a value slightly in excess of the design value.

EXPERIMENTAL RESULTS

The test results shown here cover nozzles having nominal ratios of divergence ranging from 1 to 7.9. The nominal ratio of divergence is the ratio of a nominal exit area to the throat area of the nozzle. The nominal exit area is obtained by projecting the nozzle passage to a plane normal to the passage axis at the intersection of the axis with the exit plane of the nozzle group. This method is shown quite clearly in Fig. 4 where the nominal exit area of a single passage is 0.224 × 0.604 sq in., and the throat area is 0.162 × 0.604 sq in. The nominal divergence ratio, being purely arbitrary, is used only for purposes of identification of nozzles.

The measured values of nozzle performance are shown in Figs. 3 to 7, inclusive, in terms of velocity coefficient and flow coefficient. The velocity coefficient is the ratio of the measured velocity, deduced as just explained, to the velocity for isentropic expansion from the stagnation state at inlet to the exhaust-chamber pressure. The flow coefficient is the ratio of the measured flow to the flow for isentropic expansion from the stagnation state at inlet to the exhaust pressure in the nozzle throat, or to the velocity of sound in the nozzle throat. The exhaust pressure is assumed to exist in the nozzle throat for the calculation of isentropic flow whenever this assumption yields a velocity less than the velocity of sound; otherwise the velocity of sound is assumed to exist in the nozzle throat. Another assumption which will be discussed is employed for some of the curves in Fig. 12.

Isentropic values were obtained from the Steam Tables of Keenan and Keyes.[4] The initial state for all tests shown here corresponded to a pressure of 200 psia and a superheat of 200 F. The isentrope through this state crosses the saturation line at a pressure of 16.5 psia, a pressure ratio of 0.22, and an isentropic velocity of 2586 fps. For isentropic velocities in excess of 2586 fps, therefore, some two-phase effect must be present in the measured values of velocity coefficient (but not in the measured values of flow coefficient).

Over a considerable range it is probable that expansion proceeds in the nozzle in the supersaturated state corresponding in all respects to a single-phase fluid. Nevertheless, the isentropic velocity as calculated is that for two-phase equilibrium and therefore is greater than that for the single phase. If the isentropic values had been computed for completely supersaturated, or single-phase expansion, the maximum velocity coefficients in Figs. 3 and 4 would have been unaffected, that in Fig. 5 would have been raised from 0.962 to 0.967, that in Fig. 6 from 0.961

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[3] Footnote 2, Fig. 10.

[4] "Thermodynamic Properties of Steam," by J. H. Keenan and F. G. Keyes, John Wiley & Sons, Inc., New York, N. Y., 1936.

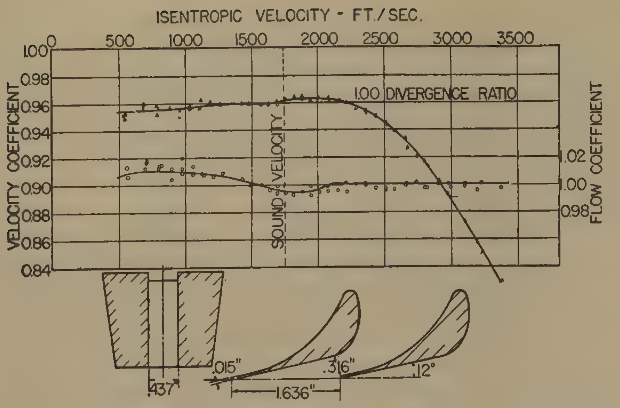

Fig. 3 Measured Coefficients for a Convergent Turbine Nozzle

(Two rows of five nozzle openings each.)

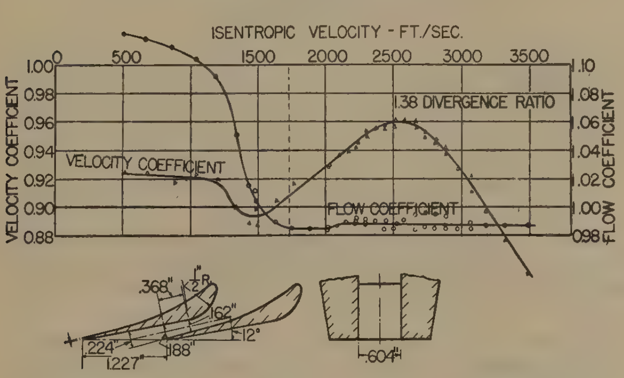

Fig. 4 Measured Coefficients for a Turbine Nozzle With a Nominal Divergence Ratio of 1.38

(Two rows of seven nozzle openings each.)

to 0.969, and that in Fig. 7 from 0.965 to unity. The last value indicates that the velocity issuing from the nozzle in Fig. 7 is augmented by condensation, but to what extent is not known.

The characteristics of the curve of velocity coefficient in Fig. 3 are similar to those for a convergent nozzle shown in the previous paper.[2] They were discussed at some length in that earlier paper.

The curve of flow coefficient in Fig. 3 shows values of about 1.01 in the subsonic range, changing to values of unity or slightly less in the supersonic range. The area of the nozzle is measured to a precision generally better than 1 per cent. During test the area may be larger than the measured area by virtue of deflection of the partitions under the pressure loading.

The high values of flow coefficient in the subsonic range are doubtless attributable to diffusion of the stream into the spaces downstream from the partition edges. This diffusion process causes the pressure in the throat to be lower than that in the exhaust chamber, and the actual mass velocity in the throat to exceed that assumed in the isentropic-flow calculation.

The isentropic mass velocity, corresponding to the velocity of sound in the throat, is greater than any value of mass velocity realized in practice. Therefore the flow coefficient should not exceed unity when this maximum value of the isentropic flow is used, that is, for pressure ratios less than 0.54. In Fig. 3 the corresponding value of the flow coefficient is 0.994 upon entering the supersonic range. The increase of about $^1/_2$ per cent as the exhaust pressure is further reduced is probably due to deflection of the nozzle partitions, and the corresponding increase in throat area under the increasing pressure differential. A similar effect may be observed in Fig. 4, and to a lesser degree in Fig. 5 where the nozzle structure is more rigid.

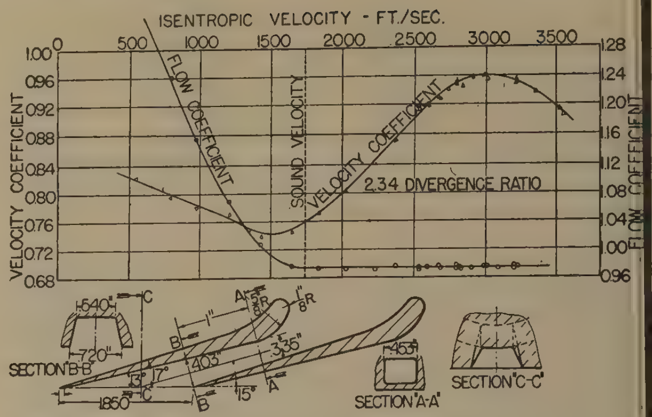

Fig. 5 Measured Coefficients for a Turbine Nozzle With a Nominal Divergence Ratio of 2.34

(Two rows of five nozzle openings each.)

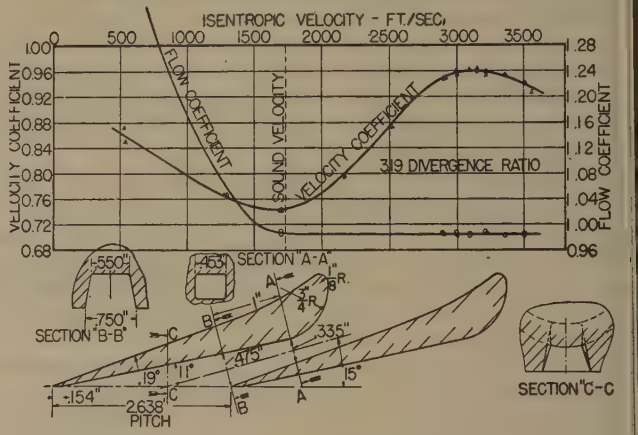

Fig. 6 Measured Coefficients for a Turbine Nozzle With a Nominal Divergence Ratio of 3.19

(Two rows of four nozzle openings each.)

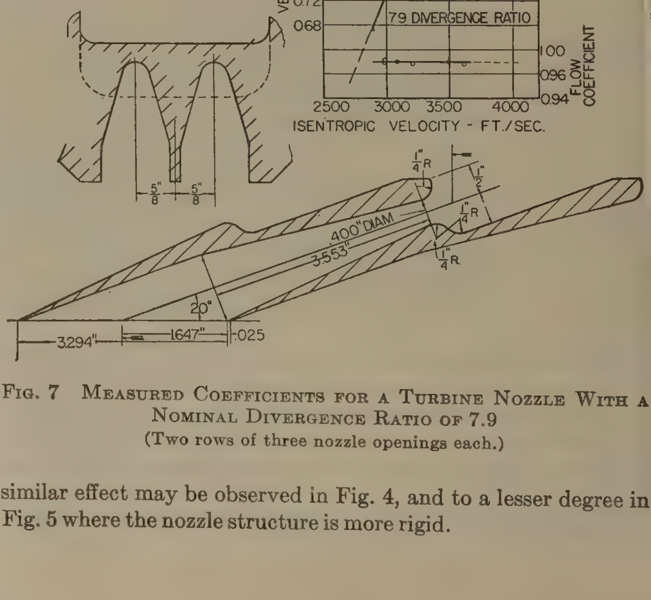

Fig. 7 Measured Coefficients for a Turbine Nozzle With a Nominal Divergence Ratio of 7.9

(Two rows of three nozzle openings each.)

The curves in Figs. 4, 5, and 6, for convergent-divergent nozzles have certain characteristics in common. These will be discussed in some detail in connection with subsequent figures.

From the nozzle cross sections accompanying these figures it will be seen that the nozzle in Fig. 4 is rectangular in cross section and has convergence and divergence in a single plane, the nozzles in Figs. 5 and 6 are rectangular in cross section and have convergence and divergence in two planes, and the nozzle in Fig. 7 is circular in cross section and is axially symmetrical up to the point where it intersects the exit plane. It follows, therefore, that the differences in performance shown by these nozzles are not solely attributable to differences in divergence ratio.

In Fig. 8 the curves of velocity coefficient for the five nozzles are compared. The curve for the largest divergence ratio is based on more experimental points in the low-velocity range than are shown in Fig. 7. The record of these additional data has been lost—only the mean curve has survived.

From Fig. 8 it may be seen that with increase in divergence ratio, a high velocity coefficient may be retained at higher and higher velocities (that is, Mach numbers). On the other hand, an increase in divergence ratio causes a decrease in velocity coefficient for any velocity less than that for maximum velocity coefficient. In the subsonic range this decrease results in extremely low values of the coefficient for high divergence ratios. For the largest divergence ratio in Fig. 8 the minimum velocity coefficient is about 0.2, which corresponds to a kinetic energy of only 4 per cent of the isentropic kinetic energy.

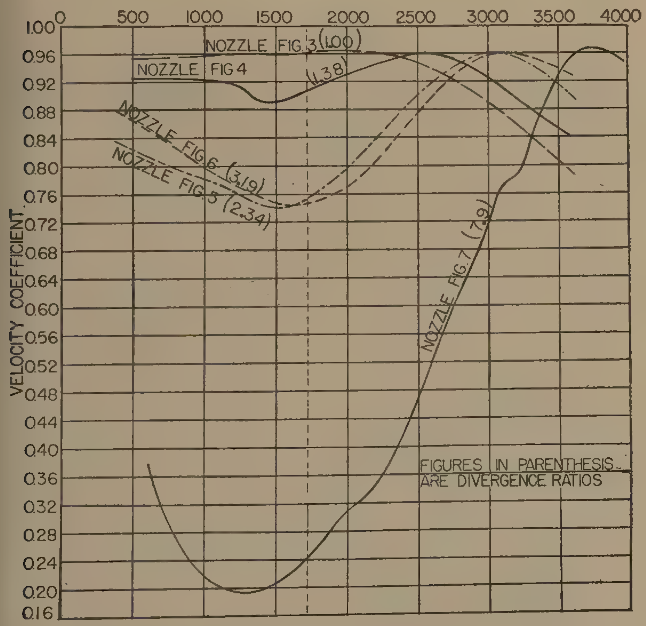

Fig. 8 Measured Velocity Coefficients for Nozzles of Various Ratios of Divergence

These characteristics may be largely accounted for through certain simple assumptions as to the processes in the convergent-divergent nozzle. Explorations of static pressure along the axis of a convergent-divergent nozzle[5] show that for high exhaust pressures the stream first expands in the nozzle to pressures below the exhaust pressure. If the exhaust pressure is high enough, the velocity in the nozzle throat will be subsonic, and the stream can compress to the exhaust pressure by simple diffusion in the divergent passage. Below a certain value of the exhaust pressure the stream expands to supersonic velocities beyond the throat and then compresses first through a shock and then by subsonic diffusion in the divergent passage.

As the exhaust pressure is lowered, the shock moves downstream. When it reaches the exit section the entire process within the nozzle is an expansion, provided that the exit section is normal to the nozzle axis and that the shock is thin. No further reduction in exhaust pressure, in the absence of boundary layer, will affect the pressure distribution in the nozzle. The change in reaction with exhaust pressure is then simply the product of the change in exhaust pressure and the exit area of the nozzle. Thus the reaction and the velocity (since the flow is unaltered by change in exhaust pressure) become linear functions of the exhaust pressure.

In Fig. 9 the uppermost curve shows the velocity for isentropic expansion plotted against the over-all pressure ratio. The broken curve *A B C D E* gives the velocity issuing from a frictionless nozzle in which the processes are as described above. From *A* to *B* the velocity in the throat is subsonic and the expansion and recompression are reversible. From *B* to *C* a shock occurs in the divergent passage, being vanishingly small in the throat at *B*, and being at its greatest in the exit section at *C*. Along *C D E*, the conditions in the nozzle follow the isentrope and remain unchanged. At *D* the exhaust pressure is identical with the lowest pressure in the nozzle passage. Between *D* and *C* the exhaust pressure is greater than this lowest pressure, and between *D* and *E* it is less.

The assumed divergence ratio for the curve *A B C D E* in Fig. 9 was selected to give coincidence with the isentrope at *D* at the pressure ratio corresponding to the maximum measured velocity coefficient. In other words, the assumed divergence ratio is determined by the pressure ratio of best performance. This procedure was followed in all subsequent calculations based upon a divergence ratio. The assumed divergence ratios are indicated on the figures.

The measured curve, which is shown in Fig. 9 for comparison, parallels the curve *C D E* at the lower exhaust pressures and the curve *A B* at the higher. In between it shows some resemblance to the curve *B C D*.

In Fig. 10 the two upper curves show the velocity coefficients

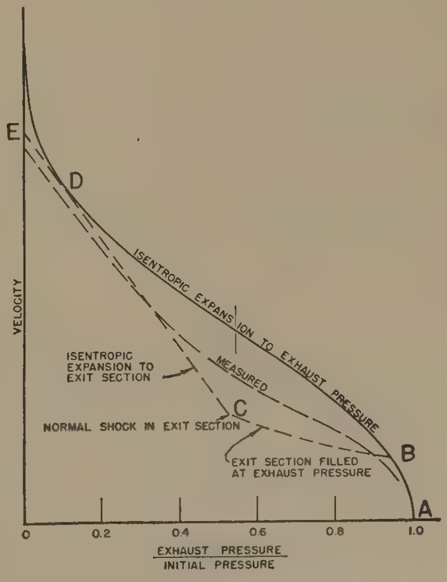

Fig. 9 Calculated and Measured Velocities
(Nominal divergence ratio 2.34, assumed divergence ratio 1.86.)

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[5] "Pressure Distribution in a Convergent-Divergent Steam Nozzle," by A. M. Binnie and M. W. Woods, Proceedings of The Institution of Mechanical Engineers, vol. 138, 1938, pp. 229–266.

corresponding to the curves in Fig. 9. The two lower curves show a similar comparison for a nominal divergence ratio of 7.9. Here the accord between measured and calculated is more striking. It appears that in this axially symmetrical nozzle the performance of the stream corresponds closely to the hypothesis, whereas in the nozzle of rectangular cross section separation occurs at the shock in the nozzle, the diffusion process is less effective, the Mach number at the shock is lower, and the loss of efficiency is correspondingly less.

In calculating the flow coefficients in Figs. 3 to 7, inclusive, the isentropic flow was based on the assumption that the exhaust pressure exists in the nozzle throat for exhaust pressures equal to or greater than that corresponding to an isentropic Mach number of unity, and that a Mach number of unity exists in the throat for lower exhaust pressures. The corresponding calculated curve is indicated in Fig. 11. As previously explained, this assumption does not even approximate reality for large divergence ratios. The calculated flow, corresponding to the assumptions of curve *A B C D E* in Fig. 9, is shown by the uppermost curve in Fig. 11, the sharp corner corresponding to point *B*. (Since Figs. 9 and 11 are for different divergence ratios, corresponding points do not lie on the same value of the abscissa.) The measured curve in Fig. 11 lies between the two calculated curves.

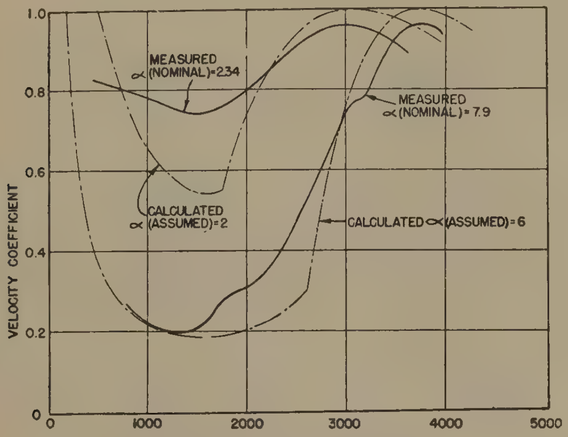

Fig. 10 Calculated and Measured Velocity Coefficients Corresponding to Fig. 9

Fig. 12 compares the flow coefficients for all five nozzles, the solid curves being based upon exhaust pressure in the throat, the dash curves on exhaust pressure in the exit section as for curve *A B C* in Fig. 9. The dash curves show an approximate constancy of coefficient over a greater range than do the solid curves. For the largest divergence ratio the second assumption fits the flow performance very well, just as it fitted the velocity performance.

For the two largest divergence ratios the point *B* in Fig. 9 occurs at exhaust pressures greater than the greatest reached in the tests. For the divergence ratio 2.34, the measurements just attained point *B*. For the divergence ratio 1.38, an appreciable part of the range *A-B* was covered by tests which gave a substantially constant flow coefficient.

The decrease in the flow, as the exhaust pressure is increased toward that of point *B*, indicates an effect of changing exhaust pressure on conditions in the nozzle throat. In view of the supersonic velocities downstream from the throat, this can be explained only in terms of transmission of the effect of exhaust pressure upstream through the boundary layer.

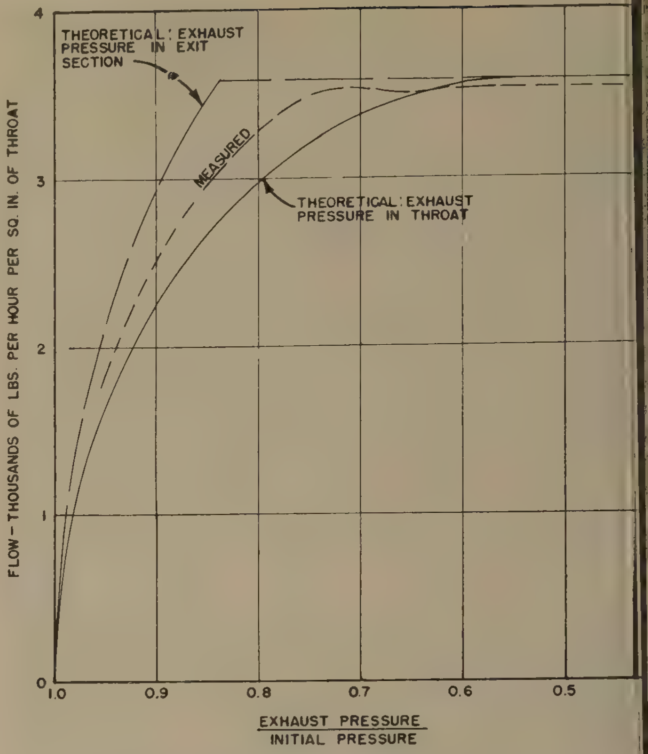

Fig. 11 Calculated and Measured Flows
(Nominal divergence ratio 1.38, assumed divergence ratio 1.3.)

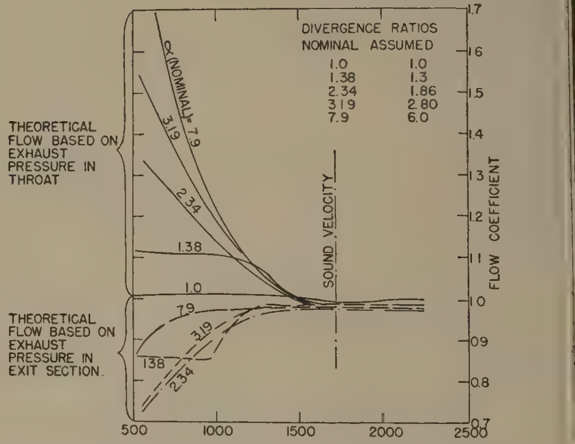

Fig. 12 Measured Flow Coefficients for Nozzles of Variou Ratios of Divergence

Conclusions

The velocity coefficient of a convergent-divergent turbin nozzle passes through a maximum at an isentropic velocity corresponding to a divergence ratio somewhat less than the nomina divergence ratio of the nozzle. It passes through a minimum a an isentropic Mach number which is less than 1, and which decreases with increase in divergence ratio.

The variation of velocity issuing from a convergent-divergen nozzle with pressure ratio has characteristics resembling more o less closely the calculated variation based on the assumption tha isentropic conditions persist in the nozzle, except for a norma shock, and that the pressure in the exit plane is identical wit the exhaust pressure whenever that is possible, and that, otherwise, the process in the nozzle passage is completely isentropic

Differences between the results of this calculation and tests can be accounted for primarily by separation at the shock, and secondarily by wall friction.

The variation of flow through a convergent-divergent nozzle is reasonably consistent with the hypothesis of the preceding paragraph. Differences between calculation and test can be accounted for by separation at the shock, transmission of exhaust-pressure effect upstream through the boundary layer, and friction, in that order of importance.

ACKNOWLEDGMENT

This paper would be incomplete without acknowledgment of the contributions to this project by Mr. Glenn B. Warren, Manager of Engineering of the Turbine Divisions of the General Electric Company. In his early days as a steam-turbine engineer, Mr. Warren conceived, designed, and constructed this testing machine. As engineer in charge of development, he continued in responsible charge of its operation, and the degree of success attained was largely due to his skill and judgment. In the early stages of its operation the machine was operated and its instrumentation was developed by Mr. Eugene E. Harris.

Discussion

D. J. Bloomberg.[6] This particular nozzle-reaction-test apparatus was developed and used to obtain comparative performance data on different types of nozzle designs and constructions, including both nonexpanding and expanding nozzles. The main objective of the test was to obtain performance data over the efficient operating range of pressure ratios.

This nozzle-reaction-test apparatus has proved to be a very reliable tool for obtaining basic flow-path data, which even up to the present time are constantly being drawn upon for design information or to assist in analyzing results on nozzle and bucket flow-path test combinations.

In elastic-fluid flow-path design work, there is a decided advantage in obtaining basic data on the separate elements comprising the entire flow path. From these data a calculation system can be worked up which can be used to predict the performance of a flow path built up of various combinations of the separate elements. The performance tests on the combination of the separate elements should serve merely as a check on the calculation system and not as a source of basic data.

The author has analyzed the test data on the expanding nozzles in the low-pressure-drop region, which is usually neglected by the practical turbine designer who is mainly interested in the good-performance region. His demonstration of a calculation system which agrees even approximately with the performance data in the low-efficiency off-design region indicates knowledge of the flow action under those conditions and should be useful either in explaining performance results or in avoiding poor flow-path conditions in flow-path design.

The author makes reference to the application of the test data, obtained approximately 20 years ago, to present-day problems involved in gas-turbine design. At the time the test data were obtained, using steam with about 200 F superheat as the elastic fluid, their application to flow paths using other elastic fluids, such as hot gases or even higher-temperature steam, was not considered. The method of presenting the data in terms of velocity corresponding to isentropic-energy drop was at that time satisfactory.

The method of presenting the test performance of a flow-path element in terms of velocity coefficient, or in terms of efficiency based on isentropic velocities or energies, as used in this paper, and also in general use in the turbine field, may be satisfactory if the data are used for flow path operating with the same elastic fluid (steam) as used in the tests and with approximately the same conditions of supersaturation, Mach number, and Reynolds number.

If the test data obtained with steam, for example, are to be used accurately for the same flow path operating with hot gases or any other elastic fluid, then the data should be presented in the form of efficiency of the actual energy available in the test flow rather than that of the theoretical isentropic energy.

A separate energy correction factor or factors should be used to correct the isentropic energies for such availability factors as supersaturation, moisture, Reynolds number, and any other factor which will affect the availability of the conversion of energy to velocity. The use of the pressure ratio instead of velocity as abscissa wil' permit a more general use of the data.

The importance of this discussion stems from the difficulty of obtaining precise basic data of flow-path performance when hot gases are the flow medium. Steam is readily obtainable and its thermodynamics properties are probably better known than those of any other flow medium. With proper interpretation, the performance data obtained with steam can be applicable to the use of any other elastic fluid.

D. L. Mordell.[7] The author's results are a beautiful illustration of the statement that a given convergent divergent nozzle is efficient only at the particular outlet Mach number corresponding to the ratio of the throat area to the discharge area.

Referring to Fig. 1 of the paper, it is observed that the velocity coefficient remains high up to an isentropic velocity of approximately 2200 fps. The writer would like to inquire whether any check of the static pressure actually existing at the vane throat was made. Since the reaction is measured, there is no means of determining whether the velocity calculated was produced in the nozzle, as a result of a contraction prior to the geometrical throat, or whether it was produced by continued expansion in the exhaust chamber. The increase in flow coefficient just past unity Mach number favors this second possibility.

If purely convergent nozzles are employed in an annulus, the flow area downstream of the blades may be greater than the throat area if the gas deflects toward the axial direction, so that supersonic velocities may be produced by a type of Prandtl-Meyer expansion around the trailing edge of the blades—the type of flow which Stodola calls expansion in the wedge. The writer's experience appears to indicate that in actual machines, expansion to Mach numbers of the order 1.4 can be attained in this manner with purely convergent nozzles. In the author's test, the high flow coefficients at such Mach number are probably due to the available room for expansion in the exhaust chamber, but in the annulus this increase in flow area must be attained by a deflection. Does the author think it might be possible to make use of this to operate at Mach numbers up to, say, 2? If so it would simplify governing problems considerably.

W. R. New.[8] Nearly as long ago as the completion of the experimental program re-examined by the author, the writer's company constructed and operated reaction testers of several designs. On these machines tests were run over a range of subsonic and supersonic velocities on a number of convergent and convergent-divergent turbine nozzles of the types commonly employed in simple impulse and velocity compounded stages. Compressed air was the fluid used, and the nozzles discharged directly into the room. The range of geometrical proportions of vane and passage cross section investigated included designs somewhat similar to the smaller divergence ratios discussed by the author. One of the significant dimensions of the nozzles tested was the relatively small height, usually $^1/_2$ or $^3/_4$ in., as frequently specified in small and moderate power steam turbines operating on high pressure.

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[6] Aircraft Gas Turbine, Engineering Division, General Electric Company, Schenectady, N. Y.

[7] Associate Professor of Mechanical Engineering, McGill University, Montreal, Can.

[8] Manager, Aviation Gas Turbine Laboratory, Westinghouse Electric Corporation, Philadelphia, Pa. Mem. ASME.

In our tests the most important characteristic of the velocity coefficient, when exhibited as a function of a Mach number, was that it reached a maximum at a Mach number greater than that corresponding to the nominal expansion ratio of the nozzle. Nominal expansion ratio was defined as the mouth area divided by the throat area of the passage. For example, in Fig. 4 of the paper we would have called $(0.188 \times 0.604) \div (0.162 \times 0.604) = 1.16$ nominal expansion ratio, whereas the author selected $(0.224 \times 0.604) \div (0.162 \times 0.604) = 1.38$ called nominal divergence ratio, to identify the nozzle. The ratio of isentropic enthalpy change at maximum velocity coefficient to that corresponding to the nominal expansion ratio of the passage ranged from about 2:1 in the case of convergent nozzles to a little over unity in the case of nozzles of large expansion ratio.

The second kind of anlytical approach employed by the author to rationalize the performance of expanding nozzles, delivering to a back pressure substantially higher than that corresponding to their expansion ratio, was found in fair agreement with the results of experiments on straight-line-axis nozzles of the type frequently used in ejectors. For turbine nozzles with a curved-line axis intersecting the plane of rotation at a very acute angle, the assumption of isentropic processes modified only by shock normal to the axis was considered to be an oversimplification.

With the focus of attention on improving the design of turbine stages rather than exploring nozzle characteristics, our use of the reaction testers for nozzle work was rather short-lived. The problems of step up, leakage, entrainment, and moisture effects on the nozzle, working in conjunction with a closely spaced bladed wheel, so dwarfed the information which could be gained by reaction tests on nozzles alone that we expanded these experimental efforts in the direction of single and multiple-stage turbine testing. Another successor to the nozzle reaction tester was the impact-traverse type of flow exploration. This we employed for many years in investigations of the effects of all features of geometry and Reynolds number in passages of larger aspect ratio, primarily those of interest in reaction-turbine stages. The exploratory traverse work was confined generally to the field of subsonic velocities.

Benjamin Pinkel.[9] The author is to be congratulated on presenting an interesting and informative paper on an important field in which there is very little information.

The following questions come to mind in reading this paper:

1 When an annular nozzle diaphragm is formed from any one of the nozzles tested as elements, what change in performance is anticipated when the flow attempts to establish radial equilibrium between the pressure and the centrifugal forces?

2 For an underexpanded nozzle, particularly in the case of the complete annular diaphragm, would one anticipate a change in direction of the flow in the free-expansion process?

Author's Closure

Mr. Bloomberg points out that "with proper interpretation, t performance data obtained with steam can be applicable to t use of any other elastic fluid"—or, one might add, to the flow steam under conditions differing from those of the tests. T effect of moisture, to which he refers, was discussed in the paper a was shown to be quite negligible in all but one of the nozzles teste

He suggests the use of pressure ratio rather than isentrop velocity for the abscissa. Probably Mach number would b better than either. A Mach number of unity is indicated in th figures.

One can readily agree with Mr. Bloomberg that the turbine d signer would like to know the effects of supersaturation, moistur Reynolds number, and (though it was not mentioned by M Bloomberg) specific-heat ratio.

In response to Mr. Mordell's question it may be said, in view various parallel investigations on nozzles similar to that of Fig. that no contraction exists prior to the geometrical throat.

It is doubtless true, as Mr. Mordell suggests, that for low e haust pressures a Prandtl-Meyer expansion occurs around th exit edge first encountered by the stream and that this results i an outward deflection of the jet (for the nozzles of.these tests a well as for an annulus of nozzles). Certainly nozzles can be de signed for Prandtl-Meyer expansion, and indeed, all oblique outlet nozzles for supersonic velocities should be so designed if th highest efficiency is the object. For a Mach number as high as such a nozzle may prove to have some divergence by virtue of th curvature of the surface near the trailing edge of the partition. I is doubtful, however, that better efficiencies will be attained at lo velocities by this method.

Several implications in Mr. New's last paragraph do not see to be well founded. The reaction method of testing can provid information on the effects of step up and of moisture. Such in formation from the reaction test is of more general applicatio than that from tests of turbine stages, because it is complicated t a lesser degree by other influences. The impact-traverse type o exploration is not simply an alternative to the reaction test. On might well arrange the impact-traverse, the reaction test, the tur bine-stage test, and the multistage turbine test, in that orde ranging from the most elementary to the most comprehensiv test. It is obvious that each has its function. A rounded turbin development program might well include all these types simul taneously—as did the program of the General Electric Company at the time these tests were run.

The answer to Mr. Pinkel's first question probably depends upon many things not specified in the question. For instance should the nozzle form be modified to make the passages "nest" together? Should the axis of the nozzle stream be directed tangent to the pitch circle of a nozzle or to that of the bucket? The problem is a complicated one to which these reaction tests yield no answer.

The answer to his second question seems to be that for a row o an annulus of nozzles if the exhaust pressure is lowered to less than that corresponding to the area ratio of the nozzle the stream will deflect outward away from the plane of the nozzle openings. In the extreme case, where the axial component of the velocity leaving the plane of the nozzle openings is everywhere supersonic, any decrease in exhaust pressure will serve to increase only the axial component of the final velocity, and will leave the tangential component unaffected.

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[9] Cleveland, Ohio.

Reaction Tests of Turbine Nozzles for Subsonic Velocities

By HANS KRAFT,[1] SCHENECTADY, N. Y.

A large number of turbine nozzles were tested by means of measuring the jet reaction. The results permit a study of the effect of most of the variables which play a role in turbine design. The design of the flow elements of turbines whether they be considered as passages or as profiles always involves a number of compromises between conflicting characteristics. Effects due to changes in the form of the nozzle partitions, their angle, their spacing, their length (called radial height), and their exit-edge thickness are shown. The tests were made during the years 1920–1928. They are here presented for the record, together with some interpretation of their meaning in the light of modern aerodynamics.

INTRODUCTION

PROF. J. H. Keenan's paper,[2] dealing with reaction tests of turbine nozzles for supersonic velocities, serves also as a fitting introduction to the present paper. There is, however, one aspect upon which this author would like to elaborate somewhat. This deals with the historical sequence of gathering knowledge on rows of nozzles or blades, or, as they are frequently referred to today in aerodynamic literature, lattices, grids, or cascades.

Of late, the gas turbine has stimulated considerable lattice research. Often one hears the opinion expressed that no consistent worth-while lattice research had been done B.G.T., i.e., before the gas turbine. It is of course largely the fault of the turbine engineer that his work during the past score of years is thus overlooked. With the exception of the English Nozzle Report[3] and the Christlein dissertation, (1)[4] not many test results have been published, although most of the large turbine builders made a notable effort to improve nozzle and blade performance. The Warren-Keenan paper (2), the Wirt paper (3), and the Kraft-Berry paper (4), describe such an effort. Professor Keenan's[2] and this paper are intended to render an account of the results obtained by Warren and Keenan.

Before the test results described here became available, the true characteristics of the velocity coefficient with spouting velocity, nozzle size, etc., were in doubt. So much so that mistaken ideas of these characteristics led to extremes in impulse-turbine design with, in some cases, both too many and too few stages. Costly mistakes could be avoided once the true characteristics were known.

It should be of interest to all concerned to see in retrospect how, without the help of a well-developed aerodynamic discipline, it was nevertheless possible to obtain improvements in performance which even today it is hard to exceed. It is a tribute to the engineer to have brought the steam turbine to a high degree of perfection without an adequate theory. What was available was test information, good judgment, correct reasoning, and a fine fingertip understanding of the pertinent factors derived from experiment.

Today's turbine nozzles are different in appearance from those presented here. The trend toward higher pressures and temperatures made them much stronger, and competitive effort bettered their performance. However, they have evolved gradually, not to say naturally, from these early specimens.

The results of these experiments will be shown in the form in which they were obtained, as will also be the conclusions drawn at that time. The only corrections which have been made are those mentioned by Professor Keenan.

A short discussion is added which attempts to explain how the results fit themselves into the modern aerodynamic picture of lattice performance.

What is the principal difference between the physical picture of 1920, and that of today? Primarily, it is the result of the modern conception of the boundary layer. While Prandtl's work was known at the time when these experiments were made, the theory of the boundary layer was not at all the finished product it is today. In fact, the important contributions of von Kármán and Pohlhausen had not yet appeared. The nozzle was considered much as a pipe in which turbulence appeared when a Reynolds number of 2000 was exceeded. The shear stress, due to viscosity, was believed to be distributed over the whole of the fluid. If the fluid stream was deflected as in a nozzle, the velocity gradient of the fluid was expected to add to the shear losses. The very important end wall losses were physically well understood as was the phenomenon of jet separation. The pressure rise along the convex wall downstream from the throat was observed, but the complicated pressure and angle distribution characteristic of the lattice was not yet known.

Today's picture of lattice flow is closer to physical reality, although a reliable and workable theory is still lacking. It is realized that the phenomena are far more complicated than was originally expected.

The flow cannot be considered as a sequence of independent jets. Rather, the flow as a whole is a continuum interrupted only by the regular sequence of the deflecting profiles. Each passage flow is under the influence of the pressure distribution established by all the other passages.

The boundary layer is the only part of the fluid affected by shear forces. It can be laminar or turbulent and different on the convex and concave side. The end wall losses appear only in the boundary layer of the end walls and "infiltrate" into the downstream flow in the form of a vortex.

The downstream stagnation point must not necessarily be at the trailing edge.

The downstream pressure distribution is seriously modified at the ends of a finite row of passages. Thus the fact that a reaction test includes the contribution of the end nozzles in its measured

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[1] Aerodynamicist, General Electric Company.

[2] "Reaction Tests of Turbine Nozzles for Supersonic Velocities," by J. H. Keenan, published in this issue of Transactions, pages 773–780.

[3] "Reports of the Steam-Nozzle Research Committee," Proceedings of The Institution of Mechanical Engineers, 1923, pp. 1–22, 311–375; 1924, pp. 455–525, 715–735; 1925, pp. 747–843; 1928, p. 31; 1930, pp. 215–315.

[4] Numbers in parentheses refer to the Bibliography at the end of the paper.

Contributed by the Power Division and presented at the Annual Meeting, New York, N. Y., November 29–December 3, 1948, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society. Paper No. 48—A-59.

performance must be considered one of the weaknesses of this type of test, especially where tests were made with more than one pair of ends. On the other hand it may be argued, as it often is, that any lattice test which does not also reproduce the time variations introduced by the rotating elements is not reliable in any case. The last word in this controversy will be spoken only when an aerodynamic theory of the running stage is available.

Nevertheless, suffice it to say that, wherever this reaction test indicated improvement, the actual turbine which incorporated the responsible design features followed suit if not always at the same magnitude.

Results of Tests

The original, individual tests of the various nozzles that were tested will not here be reproduced.

Fig. 1, which is a reproduction of Professor Keenan's Fig. 3,[2] is representative of the original form in which these test results appeared. The results are presented in various combinations which show the effect of the variables that are of importance in nozzle design. They are: form of partition (form parameter); nozzle angle; partition spacing (pitch); aspect ratio (radial height); thickness of trailing edge.

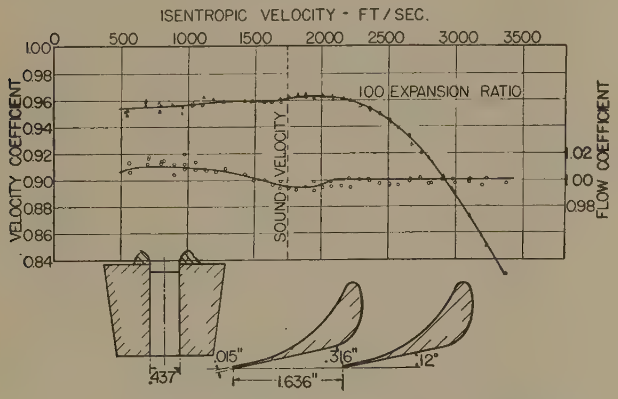

Fig. 1 Graph Representative of Original Test Results

The nozzle performance is described by velocity and flow coefficient[2] (2), which are shown as functions of isentropic velocity,[5] as determined from the variable drop from a constant initial pressure. This is essentially the picture of a Mach-number variation accompanied by a simultaneous moderate Reynolds-number change. Above sound velocity, a systematic error is introduced which is a direct consequence of the method in which the tests were made. In the supersonic range the steam issuing from the nozzle is deflected away from the nozzle face. As a result the force vector of the reaction deviates from being normal to the approach pipe.[6] The measured scale moment becomes smaller.

The supersonic-velocity coefficients shown in these tests must therefore be corrected upward by an uncertain amount. The true value is between the minimum shown here and the maximum that is obtained if a correction is made for the deflection, on the assumption that all needed stream cross section is obtained by jet deflection, neglecting crosswise expansion. This latter assumption, in many cases, will move the peak of the velocity coefficient characteristic well into the supersonic range. It is unlikely that this is correct. A good discussion of this rather complicated correction appears in Stodola's work (5).

With very few exceptions, supersonic velocities do not appear attractive for turbine design. For this reason no good correction of these supersonic coefficients has been made as yet.

Suffice it here to say that the characteristics shown in the supersonic range are increasingly conservative as velocity is increased.

If such a correction is attempted it should be noted that the center of the nozzle face is offset from the center of the pipe by approximately 2 in.

The initial condition for these tests was either 75 psia, 200 deg F superheat, or 32 psia, 250 deg F superheat. Thus both inlet conditions are within a few degrees of the same initial temperature of 500 F. It should be noted that in Figs. 3, 7, 9, and 12, the saturation line is crossed at 2890 fps. Phenomena deriving from this crossing will be dealt with in a later publication. They do not interfere with the comparisons which are presented.

A short discussion of the problems involved accompanies each comparison.

It should be borne in mind that these tests were planned and executed with the purpose of giving answers to specific design questions. The comparisons which are shown merely make the best of an existing situation. Since very few strictly consistent comparable lattices were tested, a certain amount of "reading between the lines" of the experimental data is necessary. This situation is, however, not new to the practicing engineer.

In the course of dealing with lattice information, a number of new terms were coined. The presentation is such that whenever such a term appears, it is properly explained directly in connection with the conditions that led to its creation.

Discussion of Comparisons

Form of Partition. This by itself constitutes the most important parameter, once the nozzle spacing (pitch), angle, and radial height (blade length) are given. Aerodynamically, the partition form determines to a large degree the boundary-layer behavior. By its thickness the partition displaces fluid and establishes the passage cross section; hence for continuity reasons, it controls the average fluid velocity. By prescribing a sequence of wall curvatures it can, furthermore, achieve local modifications of velocity along the wall, including the boundary layer. A partition can have a very sharp convex curvature at the throat (narrowest cross section of the passage), and thus enforce a velocity distribution along the throat which varies greatly from an average velocity. The velocity will be high at the convex wall, the boundary layer highly accelerated and thin. Downstream from the throat this velocity must drop to average velocity, and near the exit edge even below. This thickens the boundary layer and can lead to jet separation from the convex wall. In addition, the pressure distribution in the downstream space of such a nozzle will deflect the mean stream direction away from the convex partition wall. This phenomenon, sometimes called "angle exaggeration" in literal translation of a German term, calls for an over-all diffusion, i.e., pressure increase from the nozzle throat to downstream infinity. The consequence will be a high flow coefficient if the latter is based on the assumption of downstream pressure at the throat. This is the theory.

In most present-day practice not much of this diffusion will materialize. In most nozzle lattices, (and in a large number of compressor lattices also,) the exit edge cannot enforce itself as the downstream stagnation point. Then the exit angle is normally larger than theory would call for if the "Joukowsky" condition of flow separation at the end of the trailing edge were satisfied. This phenomenon leaves the determination of the stream angle to a large extent to the nonconservative shear forces in the fluid. As a result, the expected diffusion does not materialize to the degree it would if viscosity were of no consequence.

Thus there are two factors which can increase the nozzle stream angle from that given by the throat dimension: (*a*) The geometry of the contours determines the angle merely from the assumption that the stream, split by the partitions, reunites at the trailing edge (Joukowsky condition). (*b*) Fluid friction (viscosity) can modify this Joukowsky condition and make the fluid stream not follow the convex wall to the exit edge. Less of the wall is actively engaged in deflecting the stream, and the angle becomes larger.

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[5] "Thermodynamic Properties of Steam," by J. H. Keenan and F. G. Keyes, John Wiley & Sons, Inc., New York, N. Y., 1936.

[6] Refer to footnote 2, fig. 1, and reference (2), fig. 10.

The actual performance of the nozzle, as shown by test, reflects the extent to which these two factors set the flow pattern.

The effect of form can be studied in Figs. 2, 3, and 4. Fig. 2 shows a comparison of two partition forms for nearly the same nozzle pitch and angle. The radial height and with it the aspect ratio (radial height divided by throat width) is large (about 12). The 1 deg difference in nominal angle should not seriously modify the comparison. Here the higher low-velocity-flow coefficient of partition "A" reflects the sharper wall curvature at the throat. A comparison of velocity coefficients shows that below sound velocity the partition with the smaller wall curvature (larger radius) holds a slight advantage in spite of the slightly smaller nozzle angle. Above sound velocity its performance drops rapidly below that of the partition with the sharper curvature. This is a phenomenon connected with a characteristic which has been termed "negative throat." "Positive throat" would be the condition where the convex wall curvature ends, and thus the minimum passage cross section is reached before the stream passes the partition exit edge. The stream would then pass for a stretch through a passage of constant width. "Zero throat" would mean the condition where the curvature ends exactly at the throat and negative throat, where it ends downstream from the throat.

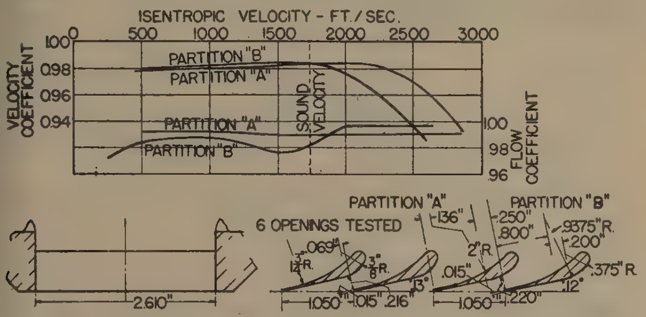

FIG. 2 COMPARISON OF TWO DIFFERENT PARTITION FORMS
(Pitch = 1.050 in.; radial height = 2.610 in.)

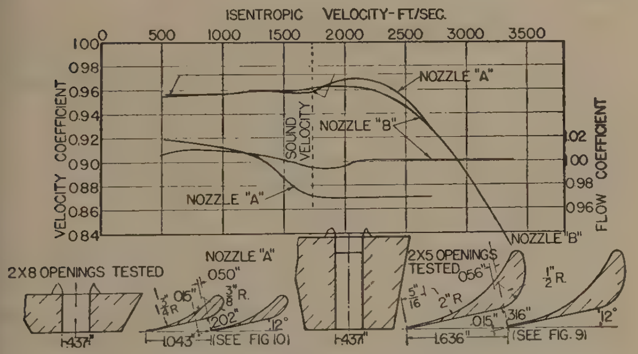

FIG. 3 COMPARISON OF TWO DIFFERENT PARTITION FORMS
(Angle 12 deg; radial height = 0.437 in.)

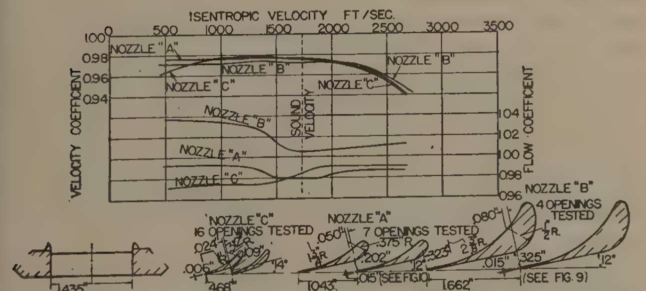

FIG. 4 COMPARISON OF THREE DIFFERENT PARTITION FORMS
(Radial height = 1.435 in.)

Experience has shown that in subsonic design negative throat is superior to positive throat. However, if such nozzles operate at supersonic velocities the situation is reversed. This is not too surprising. The supersonic-flow pattern is dependent upon the geometry of the space surface which is the locus of all points in the passage where sound velocity appears, and on the curvature characteristics of the boundaries. At the sound velocity point the wall curvature must be a maximum (6, 7). If this is not provided, and the sound velocity surface is sharply curved, it is very probable, if not certain that an isentropic flow cannot exist (6, 7). Under these conditions shocks must occur, and shocks mean energy losses. This argumentation cannot as yet be supported rigorously by theory as the art has not progressed that far. However, it does furnish an explanation of experimental fact. The slightly higher (supersonic) flow coefficient of the poorer nozzle would indicate that its sound velocity surface is larger than that of the better nozzle. It may be situated further upstream, or may be curved more sharply, or both.

Fig. 3 shows a comparison of two different nozzle designs for identical angle, radial height, and exit-edge thickness. The radial height is small. Here the pitch of the nozzles differs in addition to the form. The nozzle A with the smaller pitch should have an advantage, since its aspect ratio is 2.2, compared to the 1.4 of the larger-pitch nozzle.[7] On the other hand, the latter has a smaller exit-edge thickness in per cent of throat dimension than has the former. In the subsonic range the two nozzles are of equal efficiency. It must be concluded that these effects and the effects of Reynolds number just about balance out as regards the losses.

In the supersonic range the relatively larger edge space of the smaller-pitch nozzle becomes an advantage. It permits a better expansion behind the sound velocity surface. As very low velocities are approached, nozzle A seems to take advantage of its relatively larger negative throat. The flow coefficient of the smaller-pitch nozzle drops decisively as sound velocity is approached, to remain roughly 3 per cent lower after it has been exceeded. This again indicates that nozzle B has the larger sound velocity surface, although this difference is rather more than would be expected.

A further study of the effect of the form parameter can be made in Fig. 4. This is a comparison of three nozzles with the radial height of 1.435, and thus with rather high aspect ratios (A:7.1, B:4.4, C:13.2). A and B have the same edge thickness, thus favoring B, while the edge of C is nearly the same proportion of throat width as that of E. The angle of C is, at least nominally 2 deg larger than that of the other two nozzles, a fact which must be kept in mind when comparing performances.

As far as the velocity coefficient is concerned there is not much difference between these nozzles of such widely different design. Nozzle C, favored by its aspect ratio and somewhat larger angle, is somewhat better in the subsonic range, except for the very low-velocity region. This is significant since it certainly must be handicapped by its small scale as becomes apparent at very low velocity, where it rather quickly loses in comparison with the other nozzles.

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[7] A discussion of the advantage of high aspect ratio follows later. For the moment the statement may be accepted without proof.

Nozzle C is the only nozzle partition in all these tests in which the convex wall curvature is carried to the exit edge without a flat portion.

Nozzle B (lower aspect ratio) is poorest except above sound velocity.

The supersonic-flow coefficient of nozzle B is above 100 per cent and is rather high throughout the whole range. No good explanation for this can be advanced here especially in view of the fact that the supersonic performance is high.[8] The latter would argue against high curvature of the sound velocity surface. Nozzles A and C perform normally as far as flow coefficient is concerned. The partition with the sharper wall curvature at the throat has a higher subsonic and a lower supersonic flow coefficient.

Angle. It would seem natural that a nozzle which deflects the stream through a smaller angle should have the better performance. Less flow distortion must be introduced to obtain the desired effect. This holds good for large deflections, i.e., small nozzle angles. In the case of small deflections the local stream acceleration caused by the displacement, due to the necessarily finite partition thickness, begins to be of the order of magnitude of the acceleration needed for the useful deflection. Between the two extremes is a field where the contour of the blade is extremely important if the optimum performance is desired.

In general, the nozzle angles in these tests belong to the group where an increase in angle should be of benefit.

This is apparent in Figs. 5 and 6. There are three nozzles represented in Fig 5. All three use the same partition. The nozzle angles are 10, 13$^{1}/_{2}$, and 17 deg, respectively. Only the 10- and 17-deg nozzles are strictly comparable, both having the same pitch. The 13$^{1}/_{2}$-deg nozzle has a larger pitch. The reason for this increase is seen when the throat of the 10- and 17-deg nozzles is compared. While the 10-deg nozzle has a positive throat of 0.143 in., the 17-deg nozzle finds this dimension increased to 0.184 in. It is known that increasing the positive throat will increasingly lower the efficiency. Indeed, while the 17-deg nozzle performs better than the 10-deg nozzle, its velocity coefficient does not show as much gain as might be expected from the smaller angle through which the stream must be turned. If this same partition is spaced at a 20 per cent larger pitch, which, when set at a 13$^{1}/_{2}$-deg angle, permits a negative throat of 0.100 in., a definitely higher velocity and flow coefficient result in the subsonic range. Above sound velocity, in conformity with previous observations, the negative-throat nozzle is poorer. Also, for reasons explained before, the poorer nozzle has the higher flow coefficient above sound velocity. Both positive-throat nozzles show substantially the same uniform flow coefficient characteristic, slightly below 100 per cent.

In Fig. 6 appear two versions of nozzles using the same partition set at two different angles. Here the pitch of the larger-angle nozzle is 93 per cent of that of the smaller-angle nozzle. In consequence, while the latter has a 0.100-in. negative throat, the former has a 0.050-in. positive throat. These nozzles were tested twice. Both, especially the smaller-angle nozzle, showed what looked like an extraordinary scattering of the subsonic-velocity-coefficient readings (see Fig. 7). Over a considerable range of velocity, the points seem to group themselves along two distinct curves instead of one. Both curves are shown in Fig. 6. The test results suggest that over this range two different flow patterns can exist. From what is known now about Reynolds-number effects, this assumption cannot be entirely rejected. Indeed, similar effects, although on a smaller scale, appear in almost every nozzle test if it is run very carefully using a large number of highly accurate test points. In this case the effect is probably exaggerated by the low radial height which increases the relative proportion of fluid under shear. Also, the approach conditions peculiar to the reaction test (which are different from those of other types of test) may be partly responsible for the magnitude of these instabilities. As to the test results themselves, both nozzles show essentially the same subsonic performance, although the 18-deg nozzle really should be better. The reason for its deficiency can be found in the small pitch responsible for the positive throat. Above sound, as usual, the positive-throat nozzle at the larger angle is superior. Again, the flow coefficients of the positive-throat nozzle are lower than those with negative throats.

Partition Spacing (Pitch). The spacing of nozzle partitions can have an appreciable effect upon performance. Increasing the pitch means that the single partition must carry a higher load. The lift coefficient of nozzle partitions can be safely driven to a

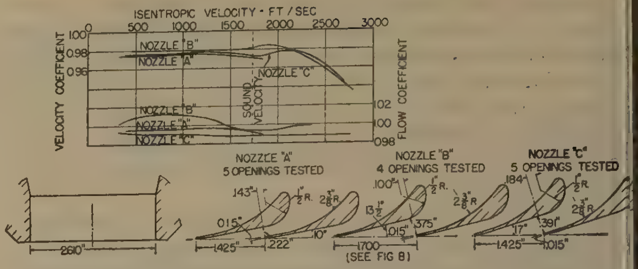

Fig. 5 Tests of Same Partition Set at Three Different Angle

(Radial height = 2.610 in.)

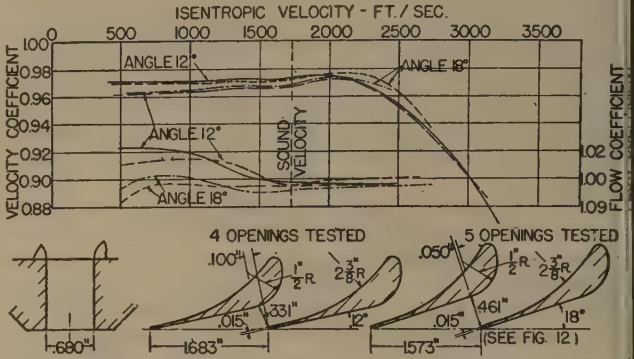

Fig. 6 Tests of Same Partition Set at Two Different Angle

(Radial height = 0.680 in.)

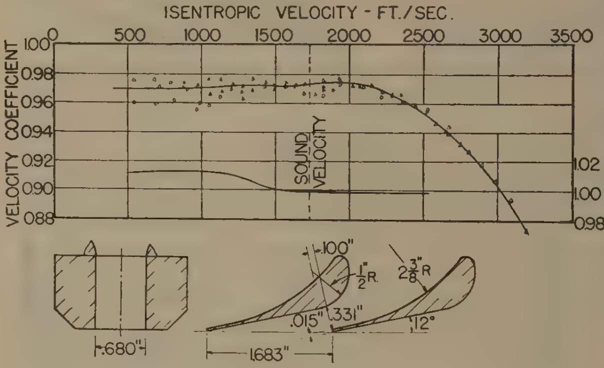

Fig. 7 Original Points of 12-Deg Angle Test in Fig. 6

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[8] Professor Keenan's remark about a possible increase in throat area under load may apply here.

multiple of the maximum lift coefficient possible on airplane profiles. Lift coefficients of 4 and higher are quite customary. In general, it can be said that for aspect ratios, which are not too small, the partition with the highest lift coefficient will furnish the highest performance. Then, in general, it will have a large negative throat and a small percentage edge thickness. When well designed, a minimum of wall surface will perform the deflection of the stream. Care must be taken in this case that the expected stream angle actually is materialized, since the "angle exaggeration" assumes noticeable proportions.

The effect of such a pitch change is apparent in Fig. 8. The wider-pitched nozzle with negative throat performs better than the nozzle with smaller pitch and positive throat. This difference persists for a good stretch into the supersonic range before it reverses to give the positive throat its high-velocity advantage. The flow-coefficient behavior shows the usual pattern, generally higher for negative throat.

Aspect Ratio. It is a well-known fact that when a fluid stream is turned, there appears along the end walls of the passages a parasitic flow pattern. The fluid that is decelerated by these end walls, i.e., their boundary layers, cannot support the centrifugal pressure gradient of the turning fluid. Some fluid material is thus short-circuited through the boundary layer, and is set into motion toward the convex partition wall. There it is deflected into the stream, and as a result, rotating fluid is carried downstream, coaxial with the main streamlines. As long as the radial height is large enough to separate the end boundary layers, this flow pattern is not affected by the height of the nozzle, its energy loss remaining the same. Thus the nozzle with the higher aspect ratio is bound to perform better. This phenomenon can be easily observed in Figs. 9, 10, and 11.

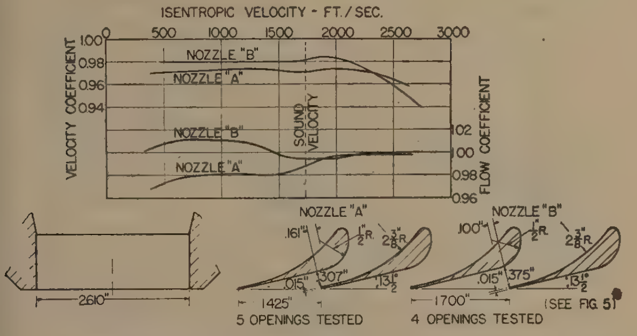

FIG. 8 TESTS OF SAME PARTITION SET AT TWO DIFFERENT PITCHES
(Radial height = 2.610; angle $13^1/_2$ deg.)

The results shown in Fig. 9 cover a variation of aspect ratio from 0.8 to 4.4 for an otherwise geometrically identical nozzle passage. There is a small change in pitch between the two models with small radial height and the third, but the difference is small enough to be neglected. The velocity coefficient behaves as must be expected. With increasing aspect ratio there is first a rapid rise, followed by a range of "diminishing return." Of somewhat more interest is the change in characteristic from 0.8 aspect ratio to 1.4. The more "temperamental" characteristic, if such a word may be applied, is a clear indication that shear forces are enforcing changes in the flow pattern as Reynolds and Mach numbers are changed. This is easily possible in passages where the parasitic end-wall flow is a noticeable part of the total flow pattern. The flow coefficient also participates in this phenomenon to a smaller extent. The high flow coefficient at low velocity of the high-aspect-ratio nozzle is worth mentioning. In all three nozzles there is clearly a diffusion after the throat which is interfered with by the end wall phenomenon. Thus it can take place less as the end-wall disturbance assumes a greater role. This physical picture fails to explain the slight decrease in low-velocity-flow coefficient from an aspect ratio of 0.8 to 1.4. Above sound velocity, the flow coefficient rises consistently with aspect ratio as might well be expected.

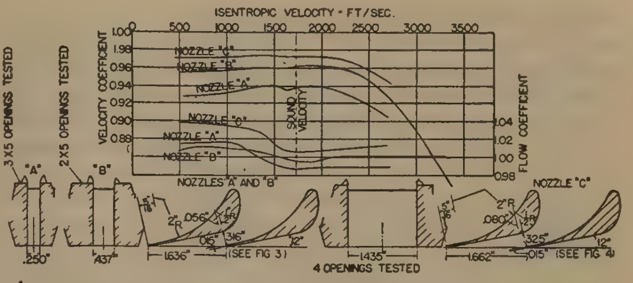

FIG. 9 TESTS OF SAME PARTITION AT THREE DIFFERENT RADIAL HEIGHTS
(Pitch = 1.65 in.; angle 12 deg.)

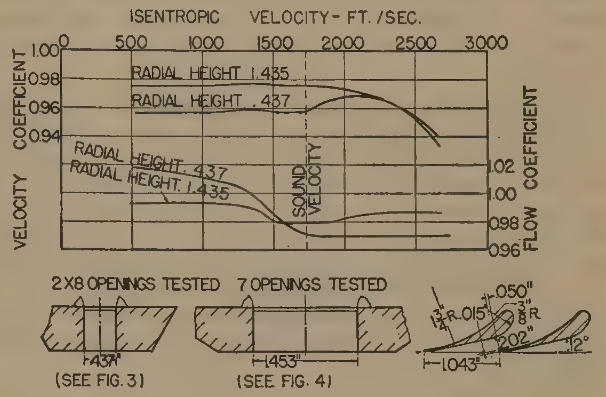

FIG. 10 TESTS OF SAME PARTITION AT TWO DIFFERENT RADIAL HEIGHTS
(Pitch = 1.043 in.; angle 12 deg.)

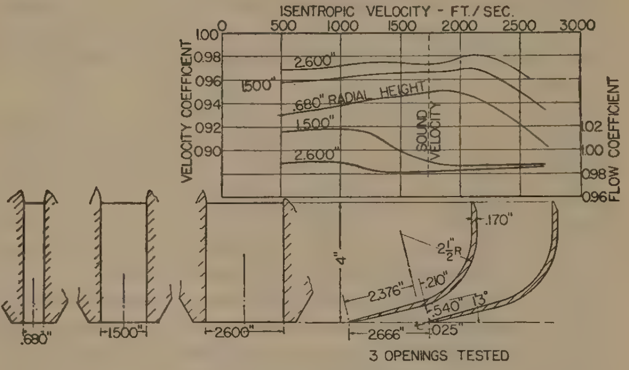

FIG. 11 TESTS OF A PLATE PARTITION AT THREE DIFFERENT RADIAL HEIGHTS
(Pitch = 2.666 in.; angle 13 deg.)

A somewhat different picture is presented in Fig. 10. Here the aspect ratios are 2.2 and 7.1. The velocity coefficient of the low-aspect-ratio nozzle increases as sound velocity is exceeded. It must be assumed that here the end-wall flow actually separates from the convex corners and that this separation is suppressed as the supersonic flow needs additional space for expansion. Such action must of course improve the performance. The large-aspect-ratio nozzle fails to show much indication of this phenomenon, although, considering its magnitude, some should be apparent if at the low aspect ratio the two end-wall patterns are really independent from each other. If this were so, the expected gain of the higher nozzles should have decreased inversely as the radial height increased. It is safe to assume that the total theoretical nozzle momentum is directly proportional to the radial height, and the loss is expressed in per cent of that theoretical momentum. The conclusion must be drawn that, in the case of the small radial height, there was interaction between the two end walls. Theoretically this is of course always the case in subsonic flow, although for large aspect ratio, it can be neglected in most cases.

In the supersonic range the velocity coefficient of the low-aspect-ratio nozzle rather early overtakes that of the other nozzle. There is much evidence that at these velocities the nozzle with the squarer throat eventually is better than that with the rectangular throat. At what Mach number one begins to be better than the other depends upon the form of the passage.

In so far as flow coefficients are concerned, that of the nozzle with the low aspect ratio appears unduly high in the subsonic zone. If, as has been done in the foregoing, a jet separation is assumed in the convex corners, then a flow coefficient above 100 per cent is not too easily explained. It calls for a diffusion from the throat downstream. A diffusion together with jet separation does not readily occur. However, it is not impossible as the pressure rise behind a concentric orifice within a pipe can demonstrate. In the supersonic range, the flow coefficient behaves normally with the poorer nozzle arriving at the higher flow coefficient.

Another comparison appears in Fig. 11. This type of nozzle partition is made by bending a beveled and sharpened flat blade. It is usually directly cast into cast-iron sidewalls. The aspect ratio varies from 1.3 to 4.8. Little need be said about the velocity-coefficient curves, except that their level is distinctly below that for formed partitions. There is very little indication that above sound velocity the squarer nozzle throat eventually will be better.

The flow coefficient of the smallest nozzle has been omitted. Its characteristic and its level indicated that there was something not in order. The other two flow coefficients are fairly consistent with the picture that has gradually become apparent. The smaller nozzle has the higher flow coefficient at low velocities. This difference practically disappears as the flow becomes supersonic. However, here the curves do not intersect.

Edge Thickness. It is easily seen that for best performance the trailing edge should have zero thickness. Since strength considerations preclude the execution of this ideal, compromises are necessary. It is important to know how much performance is lost as a safer design is used.

Fig. 12 shows two otherwise identical nozzles where the edge thicknesses are 0.015 in. and 0.050 in., respectively. The nozzles are comparatively large-angle nozzles (18 deg).

Below sound velocity there is considerable difference in performance in favor of the thinner edge. This advantage is lost as sound velocity is exceeded. The space behind the edge, which at lower velocity is filled by the turbulent wake, is utilized for expansion.

At low velocities, the thick-edged nozzle encourages diffusion after the throat, as evidenced by the high flow coefficient. Above sound velocity, the thin-edged nozzle has the slightly higher flow coefficient. This appears quite reasonable. There is no reason why the sound-velocity surfaces should differ markedly on the two nozzles. The small difference is probably caused by a small change in pressure distribution due to the different expansion into the edge space unless the thinner edge deflected measurably under the load.[2]

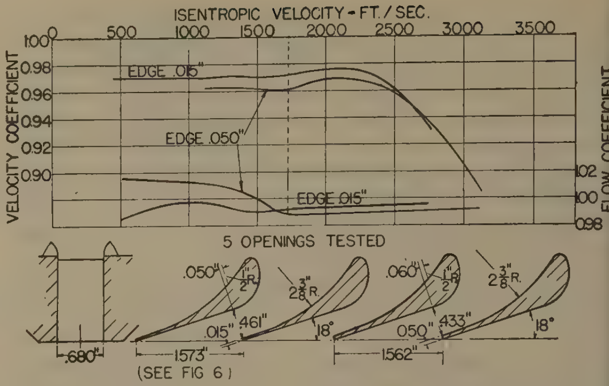

Fig. 12 Tests of Same Nozzle With Two Different Exit-Edg Dimensions
(Pitch = 1.567 in.; angle 18 deg.)

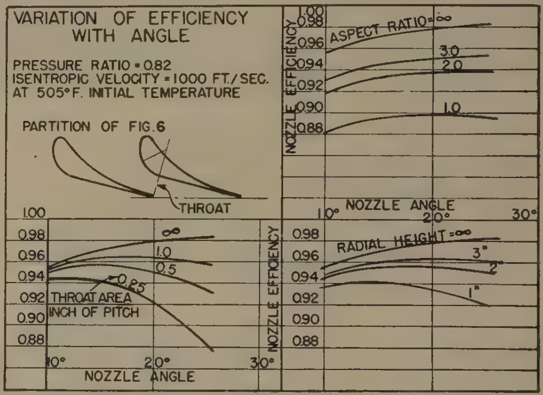

Fig. 13 Variation of Efficiency With Angle for a Formed Partition Profile

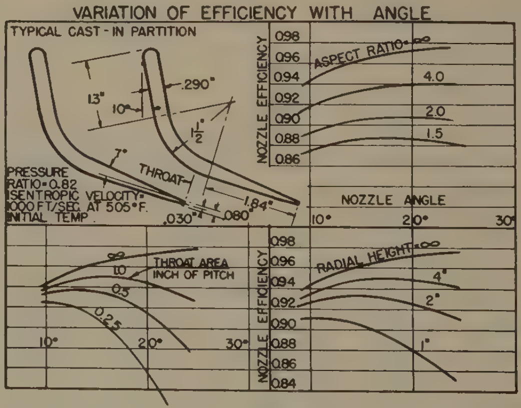

Fig. 14 Variation of Efficiency With Angle for a Plate Partition Profile

Application

Development quickly progressed past the state of knowledge provided by these experiments. It became necessary to interpolate and extrapolate beyond the regions of test information. Figs. 12 and 14 give a sample illustration of the data which thus became available.

It should be noted that these figures show efficiency, not velocity coefficient. At good performance, the efficiency is very closely equal to the square root of the velocity coefficient.

These curves fit the test information, in general, but not with-but some significant deviations. This had to be so considering the state of the art of lattice design when the experiments were made. It must, however, be emphasized again that as actual turbine experience accumulated, the conclusions drawn in these graphs were remarkably well borne out by actual turbine performance.

Conclusions

The engineer who wants to design the best possible turbine nozzle must strike the correct balance between a large number of variables:

He must weigh Mach number versus strength.

He must consider the vagaries of Reynolds number.

He must know that too high an aspect ratio will increase wall friction and edge losses; also that high negative throat is good below sound velocity and a liability above. It makes the actual stream angle and the flow coefficient uncertain. Too high a negative throat, on the other hand, will cause jet separation. Generally a larger angle is better but decreases the aspect ratio. Such an increase, if otherwise possible, can easily be overdone.

He must load his partitions as highly as possible, but must guard against losing to aspect ratio and possibly to shock losses and jet separation.

He must form his partitions such that the boundary layers represent the minimum loss.

The quantitative relationships submitted here will prove helpful in this difficult situation.

Perhaps the greatest service performed by these experiments was their catalytic action. The results, while they confirmed the physical conceptions engineers had of nozzles at that time, were quite thought-provoking in many respects. The appetite for more detailed information grew and kept growing. An impact-tube traversing test was started almost simultaneously (3). As information accumulated, the need for still more information became apparent. In the middle 1930's, this test was put on an automatic mass-production basis (4). Much has been learned since about lattice behavior. It is now known that what seemed to be inconsistencies in test results were actually caused by Reynolds-number variations due to variable approach, turbulence, wall roughness, and the like.

No matter, however, what may have been added in the meantime or whatever more will be learned in this field later, this information will retain its character as a sound and solid foundation for further progress.

Acknowledgment

Whatever Professor Keenan says about Mr. G. B. Warren's and Mr. E. E. Harris' contribution to this work wants to be reemphasized here. It was Mr. Warren who conceived and engineered this test. Mr. P. H. Knowlton, Assistant to the Manager of Engineering of the Turbine Divisions of the General Electric Company, made important contributions to the evaluation and presentation of the tests.

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The Instron Tensile Tester

By HAROLD HINDMAN[1] and G. S. BURR,[2] QUINCY, MASS.

A new type of tensile-testing instrument is described, utilizing electronic principles for both the weighing of the forces on the sample and for controlling its extension. This instrument was designed with the idea of controlling these time functions accurately, and in so far as possible, of preventing the characteristics of the machine from obscuring the accuracy of the results.

Introduction

THERE is at present a rapidly developing technology devoted to the study of those types of materials which possess complex chemical and physical structures. In this class belong principally the fields of plastics, textiles, and rubber.

It was not long ago that many of these products were not considered as being strictly engineering materials, in the sense that their physical properties could not be closely designed to suit a desired end purpose. In fact, a real difficulty existed in actually defining these physical properties in other than purely qualitative and subjective terms. Also, because formerly these materials came from natural sources, the ability to manipulate the properties to secure a desired effect was largely limited.

The situation is being changed drastically by the growth of the synthetics industry. Physical chemists are acquiring important insight into the study of long-chain molecular structure and the correlation with physical behavior. This work promises before long to enable the chemist to design on the drawing board materials which will possess an almost unlimited variety of characteristics to meet specific needs. Furthermore, a real stimulus was provided by the war toward an adequate description of already existing products in engineering language, in order that they might be produced to meet the necessary rigid specifications.

This new science has been considerably hampered by the complexity of the molecular structures involved and the attendent mechanical properties. Not only are there many indexes of behavior required, such as instantaneous modulus, recoverable and nonrecoverable creep, relaxation, and hysteresis, but these properties are strictly interdependent upon each other and upon past history with respect to time, temperature, and humidity. However, difficult as such an analysis may seem, in many cases it is just this complexity which gives to a material its most desirable characteristics. The designer has been increasingly cognizant of the correlation between mechanical behavior and service performance, and without a doubt these indexes will grow to be fundamental criteria of future material specifications.

The tensile test remains the most fundamental method of investigating the physical behavior of a material. Laboratories have long recognized that the correct evaluation of the various properties must be made against known references of humidity and temperature. In addition, it is now appreciated that, because of the predominant flow characteristics which are present, perhaps the most important parameter of all is time, which has been neglected in the past largely because of the difficulties of instrumentation. Mere ultimate tensile strength is no longer enough. The detailed shape of a stress-strain curve is assuming greater significance, and also of interest are the variations produced under more or less complex loading histories, under conditions in which the time element is rigidly controlled. By this means it is possible to develop techniques to isolate each of the various properties from one another for more detailed evaluation.

The tensile-testing instrument to be described herewith was designed in the effort to implement this new technology of polymeric materials, and to minimize the testing errors inherent in the more conventional high-inertia weighing systems. In order to satisfy the requirements of flexibility and accuracy, recent developments in electronic principles were utilized in both the load-weighing system and the crosshead control. It is hoped that its resultant versatility will find use in many other fields, such as paper, fine wire and foils, and adhesives, in which adequate testing instrumentation has so far been unavailable.

General Characteristics of New Instrument

The instrument incorporates a highly sensitive electronic weighing system employing bonded-wire strain gages for detecting and recording the load applied to the sample under test. The moving crosshead is operated by two vertical drive screws from a unique servodrive which provides an unusual flexibility of control over the motion of the pulling jaw. The chart of the recorder is driven synchronously at a wide variety of speed ratios with respect to the crosshead, thus enabling measurements of extension to be made with a large choice of magnification factors. The general characteristics of the machine may be outlined as follows:

1 By the use of readily interchangeable load cells, full scale sensitivities may be obtained extending from 2 grams up to 1000 lb or more. Therefore it is possible to test, for instance, the smallest of single fibers as well as materials having high ultimate strengths.

2 The accuracy of the load-detecting system is independent of the range in use, and is better than ±0.5 per cent.

3 The load-recording device has a high speed of response, which is of value in determining the short-time creep properties of materials as well as studying such things as multicomponent breaks. The response speed is likewise independent of the load range.

4 The load cells exhibit essentially no deflection under the applied load. Therefore the motion of the pulling jaw alone determines the rates of extension of the sample, an important factor in view of the time dependence of the properties of many types of materials. Also, a close study is possible of the rupture characteristics of even the most inextensible materials.

5 A more or less necessary corollary of the foregoing of course is the fact that the load-weighing system itself, aside from the recorder, exhibits essentially no mechanical inertia; hence it does not influence through its own action the properties of the sample to be measured.

6 Through the operation of the servocontrolled drive, a simple means is available for obtaining exact crosshead speeds over the range from 0.02 to 20 ipm. This enables precise measurements to be made of materials having widely different degrees of extensibility, as well as enabling tests on a particular sample at greatly different loading conditions.

7 Once selected, the crosshead speed remains absolutely con-

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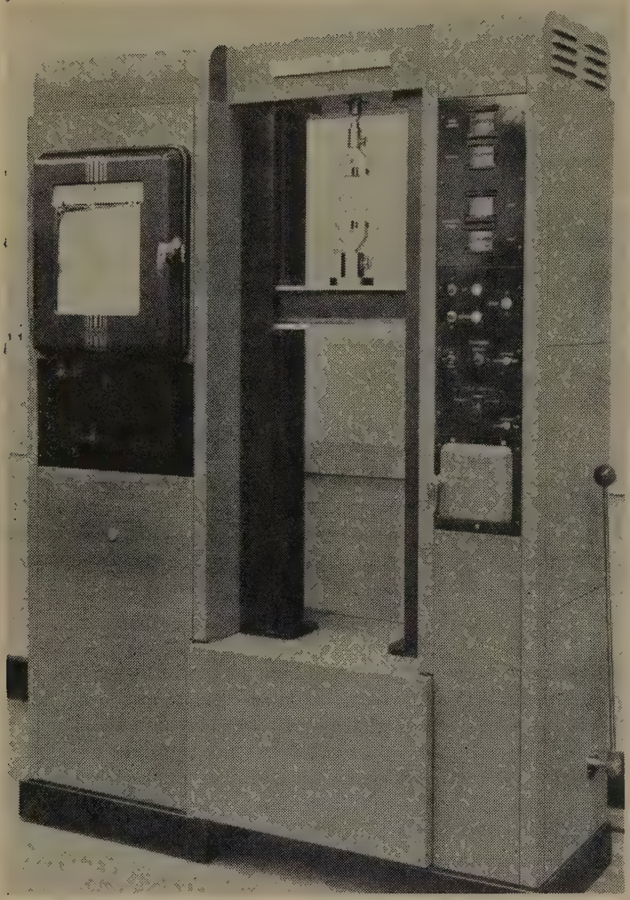


Fig. 1 An Electronic Tensile-Testing Instrument

stant for all loads up to the capacity of the machine, so that the time factor is evaluated correctly in relation to the sample characteristics.

8 The crosshead motion may be started, stopped, and reversed almost instantly, either manually or automatically. This feature enables the application of virtually any type of more or less complex loading history to the sample.

The Load-Weighing System

The load-sensitive devices are incorporated in the removable load cells, one of which is shown in Fig. 2, and which are inserted into the upper, fixed crosshead. For the lower ranges, the gages are applied to the surfaces of a bending beam, or cantilever, to which the load is applied by means of a spindle extending through the bottom of the cell. The cells for the higher ranges use a strain-sensitive member consisting of a simple tension bar with the gages applied to an area of reduced cross section.

In the design of all these cells except for the lowest range, the central load-sensitive spindle is supported rigidly by means of diaphragms, thereby eliminating sensitivity to nonaxial loading. Positive stops protect the unit against accidental overloading, and the various jaws are attached to the spindle by means of a bayonet-type coupling for ease of connection.

The strain gages are connected in the arms of a Wheatstone-bridge circuit operated from a highly stabilized 390-cycle oscillator, as shown in Fig. 3. The resulting signal is amplified by a circuit which incorporates flexible means for balancing the gages, so that the varying weights of jaws, fixtures, and the samples themselves may be compensated for. As a part of the amplifier

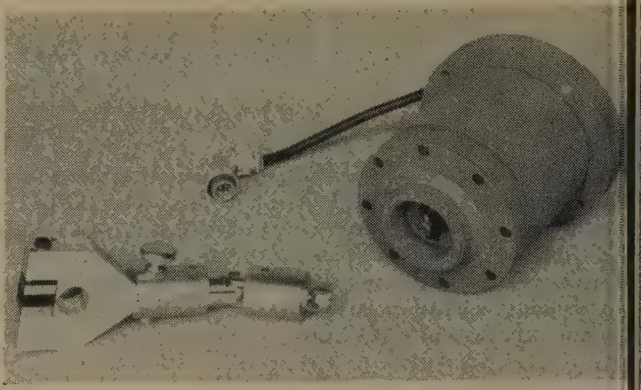
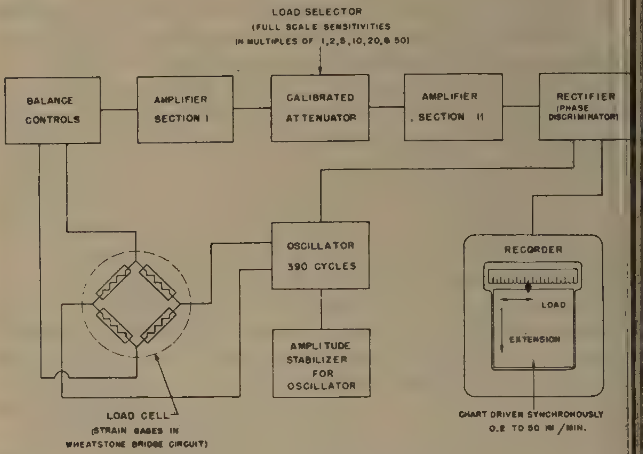


Fig. 2 Removable Load Cell



LOAD SELECTOR
(FULL SCALE SENSITIVITIES
IN MULTIPLES OF 1,2,5,10,20,& 50)
BALANCE CONTROLS
AMPLIFIER SECTION I
CALIBRATED ATTENUATOR
AMPLIFIER SECTION II
RECTIFIER (PHASE DISCRIMINATOR)
OSCILLATOR 390 CYCLES
AMPLITUDE STABILIZER FOR OSCILLATOR
RECORDER
LOAD
EXTENSION
LOAD CELL
(STRAIN GAGES IN WHEATSTONE BRIDGE CIRCUIT)
CHART DRIVEN SYNCHRONOUSLY
0.2 TO 50 IN /MIN.

Fig. 3 Load-Weighing System

is a calibrated control for changing its sensitivity in steps of 1, 5, 10, 20, and 50, thus enabling each load cell to provide a numb of full-scale load ranges. An additional control provides mea for varying the sensitivity continuously between any of the steps, so that the instrument may be calibrated for any desir odd value of full-scale load, such as grams per denier. The sign is then rectified in a special circuit which removes effectively t effects of such undesired voltages as, for example, may arise fro stray 60-cycle fields. The load signal then operates a Leeds a Northrup high-speed potentiometer recorder, into which is built small gearbox for driving the chart synchronously at spee ranging from 0.2 to 50 ipm. This portion of the instrument a the load-amplifier controls are shown in Fig. 4.

The design of the oscillator and amplifying stages is such th the sensitivity of the system remains constant over long perio of time. The instrument may be recalibrated in a few momen by hanging small weights to the upper jaw and adjusting t sensitivity, a process which is considerably simplified by the fa that the calibration need be performed for only one of the sever load ranges within the capacity of each cell to be effective for a the other ranges. Thus to check the 50-lb cell, only a 1- weight need be used to adjust the sensitivity with full sca accuracy.

Through the use of additional cells and accessories, it is possib to test in compression and flexure over the entire range of t machine while retaining its fundamental flexibility of contr Furthermore, the nature of the cells is such for certain rang that a test may be carried continuously from tension right throug into compression for special studies.

The ability of each cell to provide a multiplicity of full sca

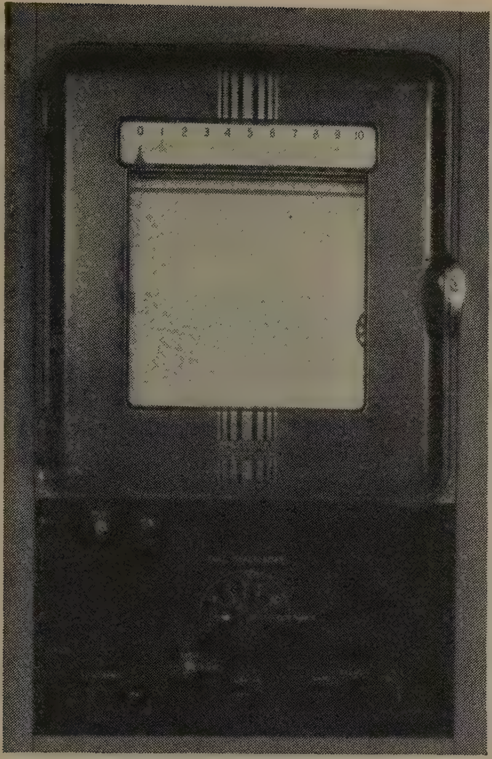


Fig. 4 High-Speed Potentiometer Recorder and Load-Amplifier Controls

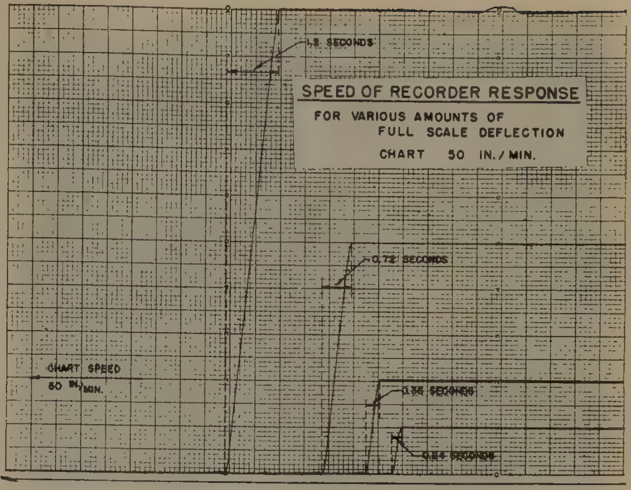


Fig. 5 Speed of Recorder Response

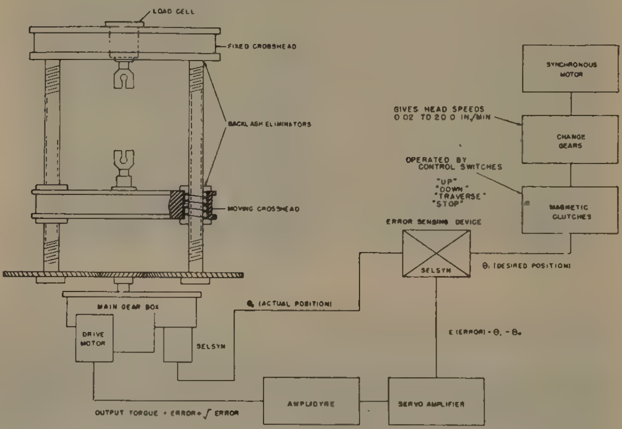


Fig. 6 Drive System
(Basic operation of positional servo.)

load ranges is limited, namely, on one hand by the desire to limit its deflection in most cases to an arbitrary value of 0.002 in., and on the other hand, by the necessity of obtaining a certain minimum electrical signal. The feature of multiple ranges has proved specially useful not only by enabling the operator to select easily the correct range for maximum clarity of record, but also by providing means of expanding by several times a certain portion of a load-elongation curve for more detailed study.

Inasmuch as the cells have a natural frequency of several hundred cycles a second, it is the recorder which places the main limitation on the speed with which rapid fluctuations of load may be followed accurately. A high speed of response is most desirable in view of the short time constants encountered in many flow characteristics, as well as for tear tests and adhesive measurements in which the load function is abruptly discontinuous. Fig. 5 is an actual determination of the response characteristics of the recorder used in this instrument. The records were obtained by applying sudden changes of input equivalent to various fractions of full scale deflection and then measuring from the chart the time to reattain equilibrium. About 1.2 sec is required for a full scale travel, and it will be observed that the response time is roughly proportional to the amount of deflection. Thus small changes in load can be followed with quite rapid response.

The Drive System

The motion of the pulling jaw is operated by means of a positionally controlled servomechanism utilizing an amplidyne drive and selsyn control elements. The operation of this system is shown schematically in Fig. 6. A follower selsyn located in the main gearbox is rotated mechanically by the drive screws so that its angular position θ_0 is a measure of the actual position of the crosshead at any instant. This unit is connected electrically to a reference selsyn, whose angular position θ_i constitutes a measure of the instantaneous desired crosshead position. A voltage may be obtained which is proportional to the difference between θ_0 and θ_i, or the error in position at any instant. The amplidyne generator is operated by this error voltage in such a way as to cause a rotation of the drive screws, thereby making the error approach zero. The torque produced by the amplidyne is proportional to the combination of both the magnitude of error and its time integral, so that automatic compensation is obtained for the effects of both a large external load or a constant applied velocity. In actual practice, the error in crosshead position is usually less than 0.0005 in. under steady-state conditions, and the system will respond very rapidly to a suddenly applied transient, such as an imposed reversal of motion.

It is now apparent that by controlling the rotation of the low-power reference selsyn, a convenient and flexible means is available for operating the motion of the pulling jaw. The reference drive consists of a small synchronous motor, a simple change-gear system for providing the desired head speeds, and fast-acting computer-type magnetic clutches for starting, stopping, and reversing the direction of rotation of the selsyn. The clutches are controlled by manual and automatic switches, Fig. 7, whose arrangement is such that almost any type of varied loading history may be applied to the test sample to suit the needs and ingenuity of the operator.

In addition to "up," "down," and "stop" buttons, "traverse" controls are provided for rapidly moving the crosshead at its

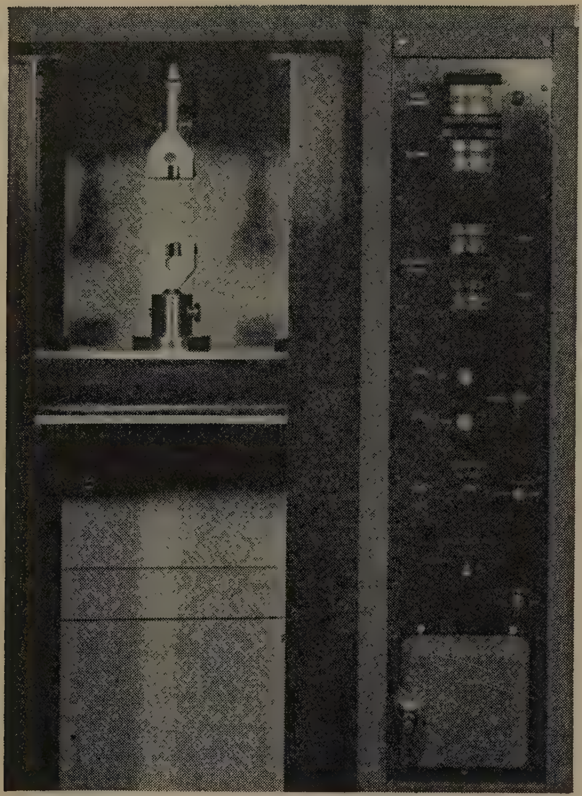


Fig. 7 Manual and Automatic Switches Control Clutches

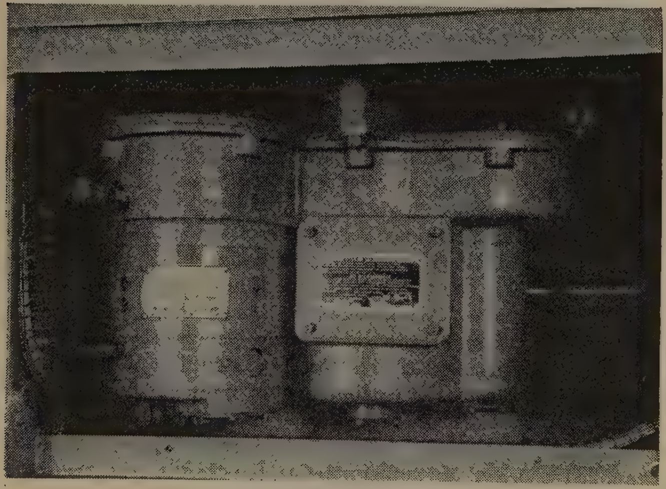


Fig. 8 Main Gearbox

maximum speed in either direction. A set of adjustable vernier dials operate additional switches to perform the following automatic functions:

1 Stop the return travel of the jaw at a previously set gage-length point to assure reproducibility in the length of sample tested.

2 Initiate rapid return of the crosshead at a point beyond the maximum extension of the sample, to speed routine testing.

3 Cycle the crosshead between two adjustable extension points for hysteresis measurements, or stop it at these points for relaxation studies.

4 Cycle the instrument between any desired load points, by means of similar controls located in the recorder.

Although easily adjustable, these vernier dials may be set to within 0.001 in. of crosshead travel. The small door, locate below these controls, houses the change-gear arrangement f determining the crosshead speeds.

Means are provided for making a mark on the chart wheneve the jaw leaves or returns to the position of initial gage lengt Therefore it is possible to evaluate correctly the amount original slack which may be present in the sample after it ha been inserted in the machine, as well as to determine such facto as permanent deformation after successive degrees of loading.

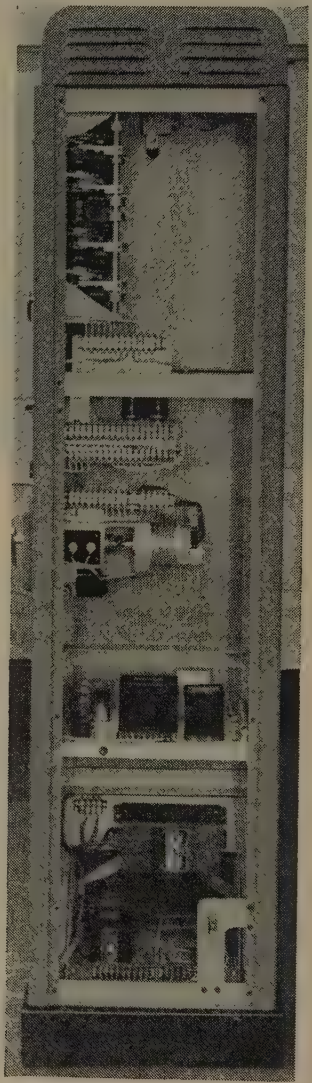


Fig. 9 Unitized Construction

The application of servodesign principles has resulted in testing instrument of a very flexible nature, but at the same tim avoiding what might seem to be a necessary complexity of the various components. The illustration of the main gearbox, Fig 8, shows the resulting compactness of mechanical design, while i Fig. 9 the unitized, accessible method of construction of the control assemblies is shown, an advantage made possible by thei primarily electrical nature. The servomethod of operation enables the addition of accessories, now under development, for producing constant rates of load and constant rates o actual strain, pending the completion of a suitable universal extensometer for measuring this quantity.

Typical Load-Elongation Records

Several actual records are herewith presented to demonstrate he type of performance which can be expected from an instrunent of this design, and its application to a variety of different aterials. Each record has been chosen to illustrate some ignificant behavior of the material, and it is believed that in cerain cases results are obtained which had previously been known nly through theoretical analysis.

Fig. 10 represents the simple load-elongation curve of a single iber of Australian 70's wool. It demonstrates the use of a highensitivity load range, and shows clearly the elastic region and low properties of the material. The spring constant of the riginal kinkiness in the fiber may be evaluated from the original ortion of the curve.

Fig. 11 illustrates the similar behavior of a single fiber of a ynthetic material, namely, viscose rayon.

Fig. 12 is the tensile curve of a sample of Fiberglas yarn. Beause of the low inherent extensibility of the yarn, a high extenion magnification ratio was used. It will be noted that, whereas he latter portion of the curve was made using a 2000-gram scale, he initial part was recorded with a 100-gram range, thus magniying the load scale for that part of the record shown dotted. In this case this technique showed the early nonlinearity as the fibers in the yarn were aligning themselves during the initial application of load. After the rupture of the sample at maximum load, the record displays the effects of interfiber friction as the specimen continues to be pulled apart.

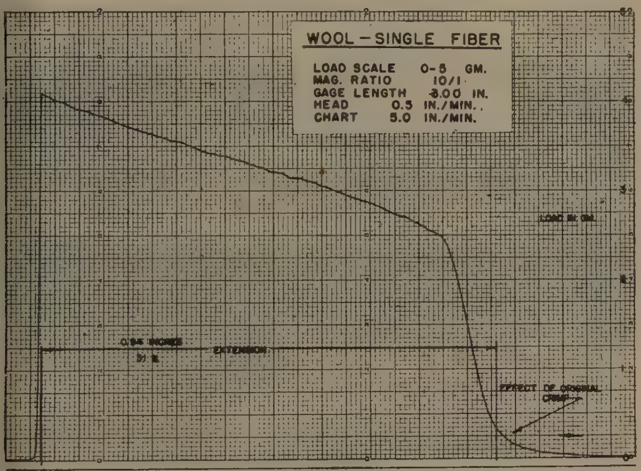


Fig. 10 Load-Elongation Curve of Single Fiber of Australian 70's Wool

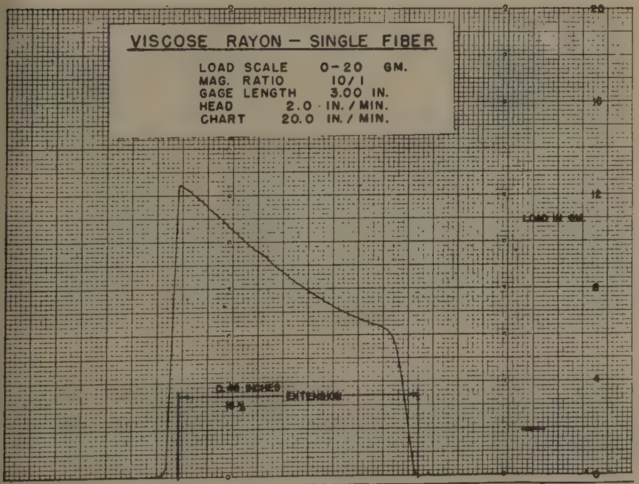


Fig. 11 Load-Elongation Curve of Single Fiber of Synthetic Material

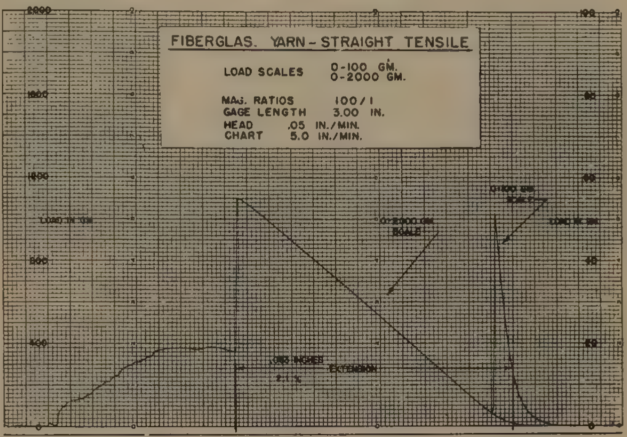


Fig. 12 Tensile Curve of Fiberglas Yarn

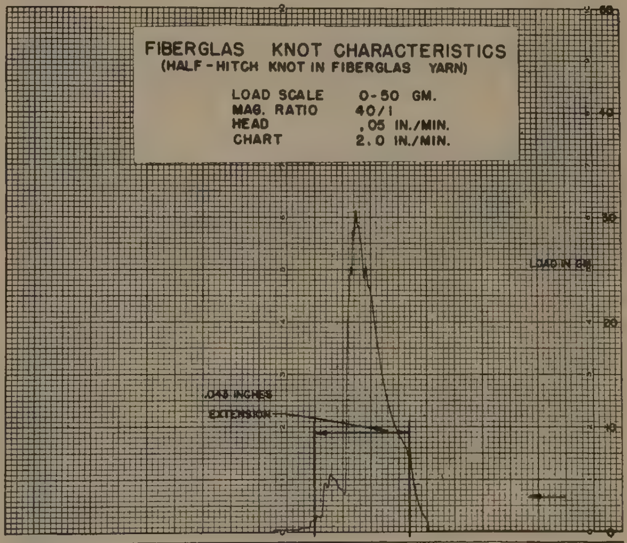


Fig. 13 Rupture Characteristics of a Half-Hitch Fiberglas Knot

Fig. 13 demonstrates the rupture characteristics of a half-hitch knot tied in a Fiberglas yarn. Each of the discontinuities in the curve represents the breaking of single fibers, or bundles of them, as the knot tightens, and shows the advantage of a load-weighing system that has low inherent deflection.

Fig. 14 shows the flow behavior of undrawn nylon yarn, which has very high ultimate extensibility. Of interest are the initial elastic region, the regression in load after the yield point has been reached, and the long flow region with a sudden discontinuity as crystallization is completed, and the subsequent formation of a new set of properties.

Fig. 15 is an example of a material of fairly high strength, and shows how the manner of rupture of a multicomponent system may be analyzed. Here the first two load changes indicate the breaking of the major plies, while the last remaining ply disintegrates fiber by fiber.

Fig. 16 shows a standard type of tear test on a fabric, and demonstrates the need for a high speed of recorder response.

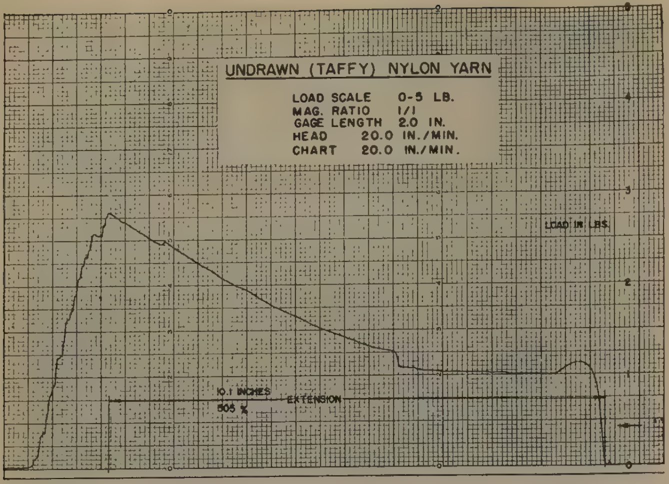


Fig. 14 Flow Behavior of Undrawn Nylon Yarn

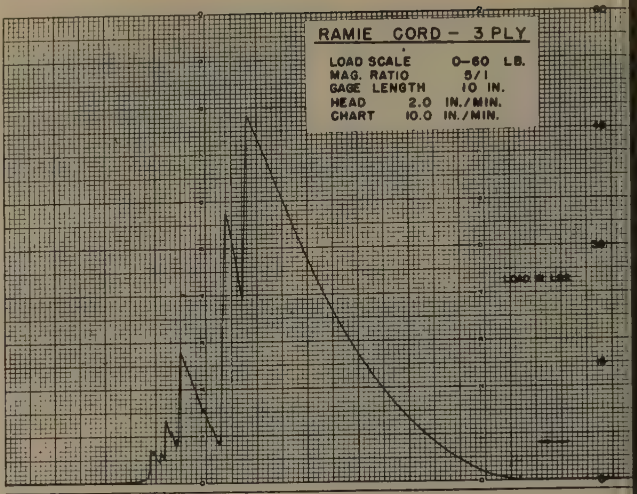


Fig. 15 Manner of Rupture of a Multicomponent System

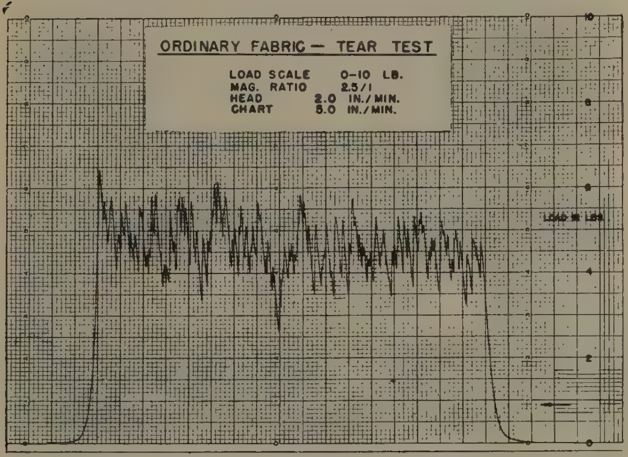


Fig. 16 Standard Tear Test

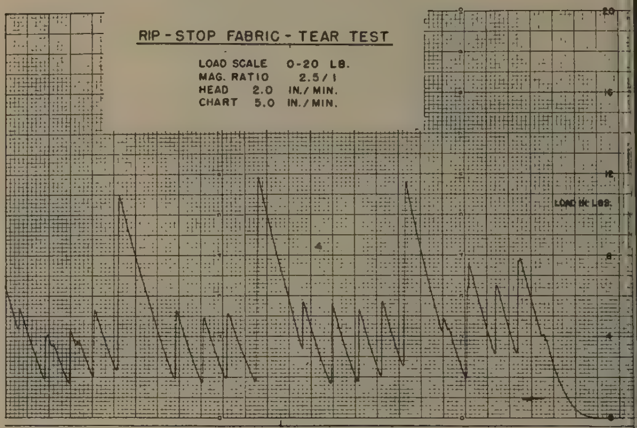


Fig. 17 Tear Test on Special Fabric

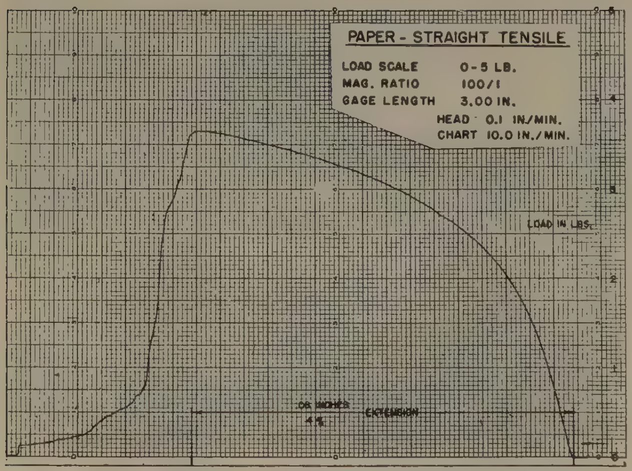


Fig. 18 Tensile Characteristics of High-Grade Paper

Fig. 17 shows the behavior of fabric specially designed to resist tearing, in which a stronger yarn is woven into the material at regular intervals as an inhibitor.

Fig. 18 illustrates the tensile characteristics of a high-grade paper, another material of low extensibility, requiring high magnification. Rupture characteristically does not take plac at once, but, instead, the individual fibers are pulled apar gradually.

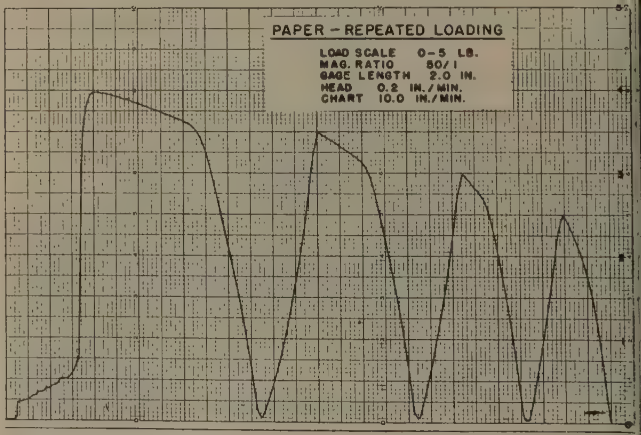


Fig. 19 Tensile Characteristics of Paper Under Successivel Higher Load Values

Fig. 19 is the same type of paper sample but there the cross-head was reversed at a number of successively higher load values the tension being released each time. This technique shows tha a form of work-hardening takes place in paper, since, for each

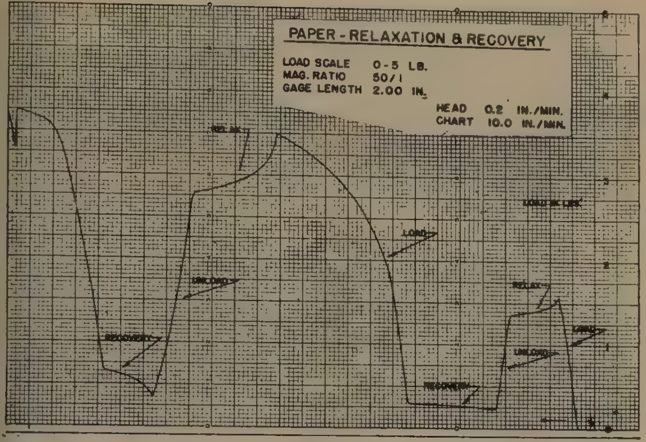


FIG. 20 RELAXATION OF LOAD WITH TIME ON PAPER SAMPLE

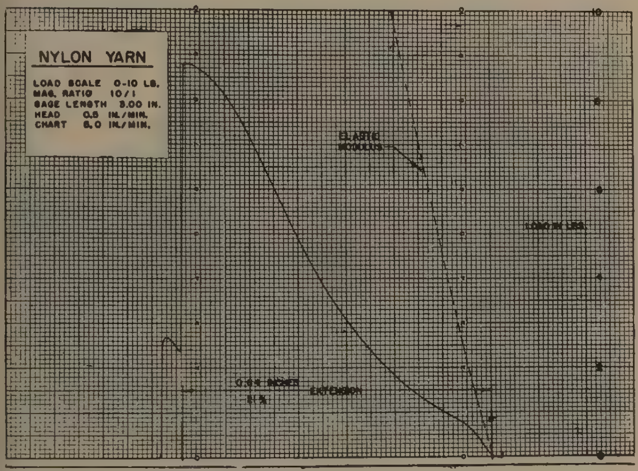


FIG. 21 LOAD-ELONGATION CURVE OF TYPICAL NYLON YARN

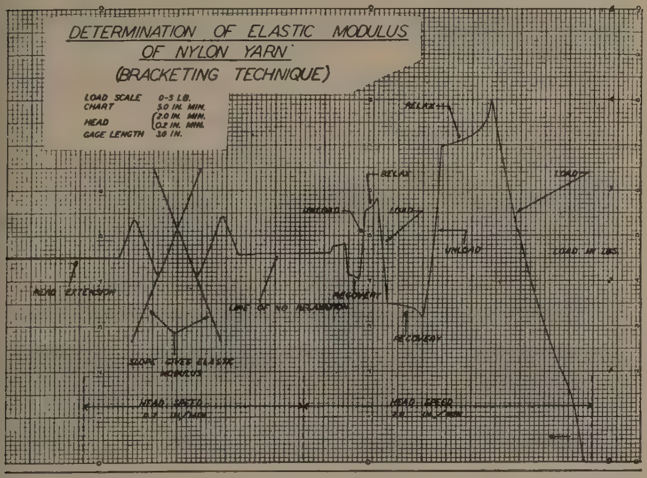


FIG. 22 METHOD OF ISOLATING MODULUS FROM OTHER PROPERTIES

cycle of loading, the flow region begins at the point of maximum previously applied load. By a slight modification of the foregoing operation, the amounts of permanent deformation may be determined as a function of varying degrees of load or extension.

In Fig. 20 the crosshead was stopped instantaneously when the load had reached a certain point. The resulting curve shows the relaxation of load with time for the given fixed extension. If now the sample is unloaded and the crosshead again stopped, the load will increase somewhat as the sample recovers from its previous conditioning. The characteristics of relaxation and recovery are shown to be much more pronounced as the sample is stretched into the flow region.

Fig. 21 is the simple load-elongation curve of a typical nylon yarn. It can be appreciated that a curve of this sort, no matter how accurate, gives very little information regarding the relative values of elastic modulus, creep, and other functions. However, in Fig. 22 is shown a method for isolating the elastic modulus from the other properties for evaluation at various degrees of extension. This technique consists of loading the sample to a certain point and then stopping the jaw motion as shown. The load will now decrease with time as relaxation takes place. Then the sample is unloaded and the jaw again stopped, at which point the load will increase with time as the sample recovers. This procedure is now repeated with decreasing increments of load, until a point is reached by a process of successive approximation, or bracketing, at which the load will not change in either direction when the jaw is stationary. As a result of the well-known "memory" effect of these materials, by which the sample "remembers" all its past conditioning, a form of dynamic equilibrium has been set up such that the relaxation of the sample is exactly counterbalancing its recovery, and temporarily only the elastic properties remain. If now the specimen is cycled over short displacements about this point of no relaxation, the slopes of the resulting records may be measured to obtain the elastic modulus. At this point the head speed was reduced by 1/10 in order to give a measurable slope to the line.

Inasmuch as the elastic modulus in most materials varies with the amount of extension of the sample, among other factors, this procedure should be repeated at successively higher extensions. The analysis will produce a graph of modulus as a function of extension similar to Fig. 23. If these values are transposed to the original load-elongation diagram, Fig. 21, a comparison may be made of the actual behavior of a material as against its hypothetical purely elastic properties.

Fig. 24 illustrates some of the possible techniques available through the use of the automatic cycling features of the instrument. In the first part of the record, the specimen was cycled between two fixed values of extension. It can be seen how the load decreases from peak to peak in a fashion similar to a pure relaxation curve. The second part of the record is an example of compound cycling, in which the sample is cycled between load at one limit and extension at the other. In this case, the limits consisted of maximum load of 3 lb, and a return to the original gage

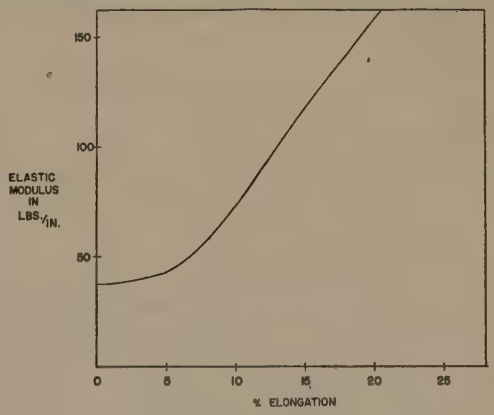


FIG. 23 MODULUS AS A FUNCTION OF PER CENT ELONGATION FOR NYLON YARN

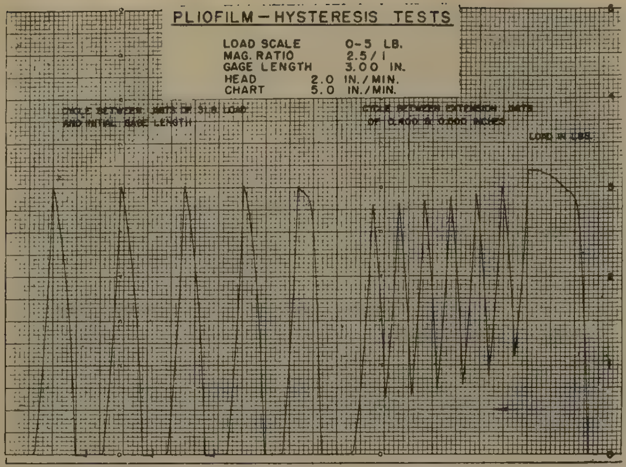


Fig. 24 Demonstrating Automatic Cycling Feature of Instrument

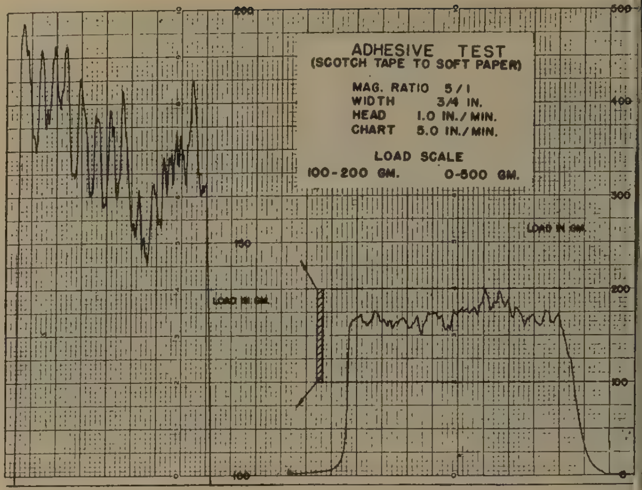


Fig. 25 Results of an Adhesive Test With Scotch Tape

length. Variations of this procedure are useful in determining such factors as work recovery, and the relations between primary and secondary creep.

Fig. 25 demonstrates an adhesive test, in which a length of Scotch tape is slowly pulled away from a sample of paper. Random fluctuations of load are obtained as the adhesive separates from or pulls out the fibers in the paper. This record also shows the manner in which the zero of the load scale may be suppressed and a certain portion expanded for more detailed observation of a record.

Fig. 26 was first obtained accidentally during an attempt to obtain the previous record. The piece of Scotch tape had been unthinkingly applied to a length of regular chart paper, and the regularly spaced dips in the curve that resulted were at first a source of some puzzlement. Further study showed, however, that these dips occurred whenever the line of separation of this tape encountered each of the fine rulings in the chart, with a somewhat broader indication occurring at the heavier division. This may be verified readily by the reader directly from the record, taking into account the magnification ratio. Thus the adhesive action was definitely sensitive to the presence of the ink in the chart rulings.

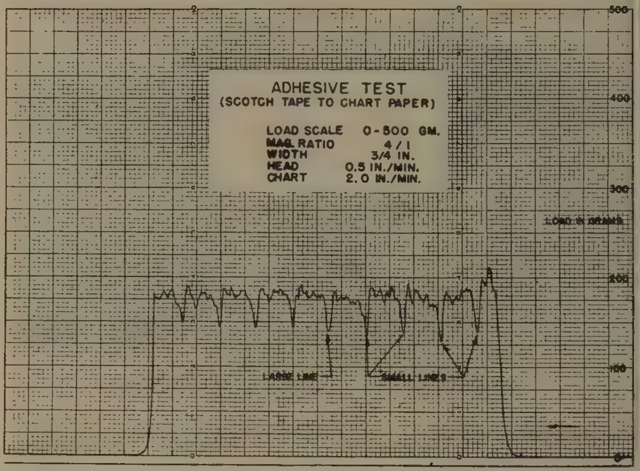


Fig. 26 Adhesive Test of Scotch Tape on Ink-Ruled Paper

Heat Transfer to Liquid Metals Flowing in Asymmetrically Heated Channels

By W. B. HARRISON[1] and J. R. MENKE[2]

This paper is a brief extension utilizing basic ideas presented in an earlier paper by R. C. Martinelli (1).[3] Heat transfer to a fluid in turbulent flow between two plane parallel walls is discussed for the case of an asymmetrically heated system in which the entire heat flux is transferred at one of the walls and no heat passes through the other. An annulus for which the ratio of diameters approaches unity may approximate this case in the limit. A few remarks are made regarding experiments which might yield information on the variation of eddy diffusivity near the center line of a channel, and a method is presented for predicting heat-transfer coefficients for asymmetrical heat-transfer systems. Since the same assumptions used by Martinelli are applied in this paper, and since the treatment is similar, this work is considered as an extension only, and it is expected that the reader should have a working knowledge of Martinelli's paper in order to follow the ideas presented here.

Nomenclature

The following nomenclature is used in the paper:

N_{Nu} = Nusselt number $= \dfrac{hD_e}{k}$

N_{Pe} = Peclet number $N_{Re} \cdot N_{Pr} = \dfrac{D_e u_m \gamma c_p}{k}$

N_{Pr} = Prandtl number $= \dfrac{c_p \mu}{k}$

N_{Re} = Reynolds number $= \dfrac{D_e u_m \gamma}{\mu}$

ln = natural logarithm to the base e
A = heat-transfer surface area, sq ft
D_e = hydraulic diameter of channel, ft

$Q = \dfrac{q_w}{\alpha c_p \gamma A_w \sqrt{\tau_w/\rho}}$

c_p = heat capacity at constant pressure, Btu/lb deg F
f = Fanning friction factor (this is the only significant deviation from nomenclature used by Martinelli)
h = heat-transfer coefficient, Btu/hr ft^2 deg F
k = thermal conductivity of fluid, Btu/hr ft^2 (deg F/ft)
q = rate of heat flow, Btu per hr
t = temperature, deg F
u = velocity of fluid, fph
y = perpendicular distance from channel wall to point in fluid stream, ft

$y^+ = y \dfrac{\sqrt{\tau_w/\rho}}{\nu} = (y/b) \dfrac{N_{Re}\sqrt{f/2}}{4}$ for parallel wall systems

$\alpha = \dfrac{\epsilon_H}{\epsilon_M}$

γ = density of coolant, lb per cu ft
ϵ = eddy diffusivity, sq ft per hr
μ = viscosity of fluid, lb/ft hr
ν = kinematic viscosity, $\dfrac{\mu}{\gamma}$ sq ft per hr
ρ = mass density of coolant, lb-hr^2/ft^4
τ = shear, psf

Subscripts:

asym = asymmetrical system
m = mean condition
max = maximum condition
sym = symmetrical system
w = refers to wall of channel
H = refers to heat
M = refers to momentum
1 = refers to t and y/b at $y^+ = 5$
2 = refers to t and y/b at $y^+ = 30$
3 = refers to t and y/b at $y^+ > 30$

Statement of the Problem

A fluid flows between two plane parallel walls, w and w'. All heat to or from the fluid stream passes through wall w. No heat flows through wall w'.

The walls are separated by a distance $2b$; the linear velocity (parallel to the walls) of the fluid at a given point is u; the distance from wall w to a given point is y. It is assumed that the hydrodynamic and thermal fields are fully established, and it is postulated that the properties of the fluid are independent of temperature.

The time variable has been neglected, and it has been assumed that, in the turbulent stream, the conditions described are reasonable averages with respect to time of the true instantaneous values.

Utilizing the analogy between momentum transfer and heat transfer, one can predict heat-transfer coefficients if velocity distribution in the channel is known. It is the problem, then, to predict the relation between Nusselt number and Reynolds number for fluids having various Prandtl numbers. Since the true values of the eddy diffusivity near the center line of a channel are not well established, the effect of different eddy-diffusivity values on the theoretical heat-transfer predictions for this case of asymmetrical heating will be examined.

Theoretical Approach to the Problem

By definition, the heat-transfer coefficient is

$$h = (q/A)_w \frac{1}{t_w - t_m} \qquad [1]$$

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[3] Numbers in parentheses refer to the Bibliography at the end of the paper.

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society. Paper No. 48—A-51.

where t_w is the wall temperature and t_m is the mixed mean fluid temperature at the same cross section.

But

$$(q/A)_w = -k\left(\frac{dt}{dy}\right)_{y=0} \quad\ldots\ldots\ldots\ldots\ldots [2]$$

where k is the thermal conductivity of the fluid at the wall.

Then

$$\frac{h}{k} = -\frac{\left(\frac{dt}{dy}\right)_{y=0}}{t_w - t_m} \quad\ldots\ldots\ldots\ldots\ldots [3]$$

and

$$N_{Nu} = \frac{hD_e}{k} = \frac{4hb}{k} = 4\left[\frac{-\frac{dt}{d(y/b)}}{t_w - t_m}\right]_{y/b=0} \quad\ldots\ldots [4]$$

Similarly to the development in Martinelli's paper, y/b and eddy diffusivity of heat ϵ_H, may be assumed equal to zero in the laminar region near the wall.

It follows that

$$-dt = \frac{q_w}{c_p\gamma A_w}\cdot\frac{N_{Pr}}{\nu}\,dy \quad\ldots\ldots\ldots\ldots\ldots [5]$$

$$-dt = \frac{q_w}{\alpha c_p\gamma A_w\sqrt{\tau_w/\rho}}\cdot\alpha\,\frac{4bu_m\gamma}{\mu}\cdot N_{Pr}\cdot\frac{\sqrt{f/2}}{4}\,d(y/b)\ldots [5a]$$

Buffer layer ($y^+ = 5$ to $y^+ = 30$)

$$-\int_{t_1}^{t_2} dt = 5Q\ln(1 + 5\alpha N_{Pr})\ldots\ldots\ldots\ldots [8$$

Turbulent core ($y^+ = 30$ to any $y_3/b > y_2/b$ @ $y^+ = 30$)

$$-\int_{t_2}^{t_3} dt = Q\int_{y_2/b}^{y_3/b}\frac{(1 - y/2b)}{f(y/b) + \dfrac{4}{\alpha N_{Pe}\sqrt{f/2}}}\,d(y/b)\ldots [9$$

$$f(y/b) \equiv \frac{\epsilon_M}{b\sqrt{\tau_w/\rho}}$$

Thus the temperature drop from the wall w to any point in th stream farther from the wall than y_2/b @ $y^+ = 30$ is

$$t_w - t_3 = 5Q\alpha N_{Pr} + 5Q\ln(1 + 5\alpha N_{Pr}) + Q\int_{y_2/b}^{y_3/b}\frac{(1 - y/2b)}{f(y/b) + \dfrac{4}{\alpha N_{Pe}\sqrt{f/2}}}\,d(y/b)\ldots [10$$

And, by definition

$$t_w - t_m \equiv \frac{\displaystyle\int_0^2 \frac{u}{u_{\max}}(t_w - t_3)\,d(y/b)}{\displaystyle\int_0^2 \frac{u}{u_{\max}}\,d(y/b)}\ldots\ldots\ldots [11$$

Then

$$N_{Nu(\text{asym})} = \frac{\alpha N_{Pe}\sqrt{f/2}\displaystyle\int_0^2 \frac{u}{u_{\max}}\,d(y/b)}{[5\alpha N_{Pr} + 5\ln(1 + 5\alpha N_{Pr})]\displaystyle\int_0^2\frac{u}{u_{\max}}\,d(y/b) + \int_{y_2/b}^2\frac{u}{u_{\max}}\int_{y_2/b}^{y_3/b}\frac{(1 - y/2b)}{f(y/b) + \dfrac{4}{\alpha N_{Pe}\sqrt{f/2}}}\,d(y/b)d(y_3/b)}\ldots [12$$

But

$$\frac{q_w}{\alpha c_p\gamma A_w\sqrt{\tau_w/\rho}} = Q$$

Then

$$-\left[\frac{dt}{d(y/b)}\right]_{y/b=0} = \frac{Q\alpha N_{Re}N_{Pr}\sqrt{f/2}}{4} = \frac{\alpha Q N_{Pe}\sqrt{f/2}}{4}\ldots [6]$$

The preceding equations hold for both the asymmetrical and the symmetrical cases.

Assuming that the laminar layer extends from $y^+ = 0$ to $y^+ = 5$

$$y^+ \equiv y\,\frac{\sqrt{\tau_w/\rho}}{\nu};$$

that the buffer layer extends from $y^+ = 5$ to $y^+ = 30$; and that the turbulent core extends from $y^+ = 30$ across the channel, the following temperature-drop equations were developed for the asymmetrical case:

Laminar layer ($y^+ = 0$ to $y^+ = 5$)

$$-\int_{t_w}^{t_1} dt = 5Q\alpha N_{Pr}\ldots\ldots\ldots\ldots\ldots [7]$$

The treatment for the symmetrical case is identical to the treatment of the asymmetrical case except that q/A is zero at the center line of the channel rather than at wall w'.

Then, for the turbulent core

$$-\int_{t_2}^{t_3} dt = Q\int_{y_2/b}^{y_3/b}\frac{(1 - y/b)}{f(y/b) + \dfrac{4}{\alpha N_{Pe}\sqrt{f/2}}}\,d(y/b)\ldots [13]$$

So

$$t_w - t_3 = 5Q\alpha N_{Pr} + 5Q\ln(1 + 5\alpha N_{Pr}) + Q\int_{y_2/b}^{y_3/b}\frac{(1 - y/b)}{f(y/b) + \dfrac{4}{\alpha N_{Pe}\sqrt{f/2}}}\,d(y/b)\ldots [14]$$

and

$$t_w - t_m = \frac{\displaystyle\int_0^1 \frac{u}{u_{\max}}(t_w - t_3)\,d(y/b)}{\displaystyle\int_0^1 \frac{u}{u_{\max}}\,d(y/b)}\ldots\ldots\ldots [15]$$

It follows that

$$N_{Nu(\text{sym})} = \frac{\alpha N_{Pe}\sqrt{f/2}\displaystyle\int_0^1 \frac{u}{u_{\max}}\,d(y/b)}{[5\alpha N_{Pr} + 5\ln(1 + 5\alpha N_{Pr})]\displaystyle\int_0^1\frac{u}{u_{\max}}\,d(y/b) + \int_{y_2/b}^1\frac{u}{u_{\max}}\int_{y_2/b}^{y_3/b}\frac{(1 - y/b)}{f(y/b) + \dfrac{4}{\alpha N_{Pe}\sqrt{f/2}}}\,d(y/b)\,d(y_3b)}\ldots [16]$$

In comparing symmetrical and asymmetrical systems, having the same distance between walls, the same Reynolds number, and the same Prandtl number, it is seen that the ratio of Nusselt numbers is as follows

$$\frac{N_{Nu(\mathrm{asym})}}{N_{Nu(\mathrm{sym})}} = \frac{2[5\alpha N_{Pr} + 5\ln(1 + 5\alpha N_{Pr})]\displaystyle\int_0^1 \frac{u}{u_{\max}}\, d(y/b) + 2\int_{y_2/b}^{1} \frac{u}{u_{\max}} \int_{y_2/b}^{y_3/b} \frac{(1 - y/b)}{f(y/b) + \dfrac{4}{\alpha N_{Pe}\sqrt{f/2}}}\, d(y/b)\, d(y_3/b)}{[5\alpha N_{Pr} + 5\ln(1 + 5\alpha N_{Pr})]\displaystyle\int_0^2 \frac{u}{u_{\max}}\, d(y/b) + \int_{y_2/b}^{2} \frac{u}{u_{\max}} \int_{y_2/b}^{y_3/b} \frac{(1 - y/2b)}{f(y/b) + \dfrac{4}{\alpha N_{Pe}\sqrt{f/2}}}\, d(y/b)\, d(y_3/b)} \quad \ldots . [17]$$

Since

$$2\int_0^1 \frac{u}{u_{\max}}\, d(y/b) = \int_0^2 \frac{u}{u_{\max}}\, d(y/b)$$

the first part of the numerator in the right-hand side of the equation is equal to the first part of the denominator. The difference in the second parts of the numerator and denominator results from different boundary conditions, that is, heat flux is assumed to be zero at the center line of the channel in the symmetrical system, but it is zero at wall w' in the asymmetrical system.

CALCULATIONS

Data of Nikuradse (2) were used to evaluate

$$\frac{u}{u_{\max}} \quad \text{and} \quad \frac{\epsilon_M}{b\sqrt{\tau_w/\rho}}$$

and these functions of y/b are shown in Figs. 1 and 2. The data were taken at $N_{Re} = 1.34 \times 10^5$, and the functions were assumed to remain the same for all Reynolds numbers between 10^4 and 10^6. The values of

$$\frac{\epsilon_M}{b\sqrt{\tau_w/\rho}}$$

determined experimentally are referred to as $f(y/b)$. Using $f(y/b)$, Nusselt numbers were calculated for the following asymmetrical cases:

$N_{Re} = 10^6$; $N_{Pr} = 0.005, 0.01, 0.05, 0.70$

$N_{Re} = 10^5$; $N_{Pr} = 0.005, 0.01, 0.05$

$N_{Re} = 10^4$; $N_{Pr} = 0.01$

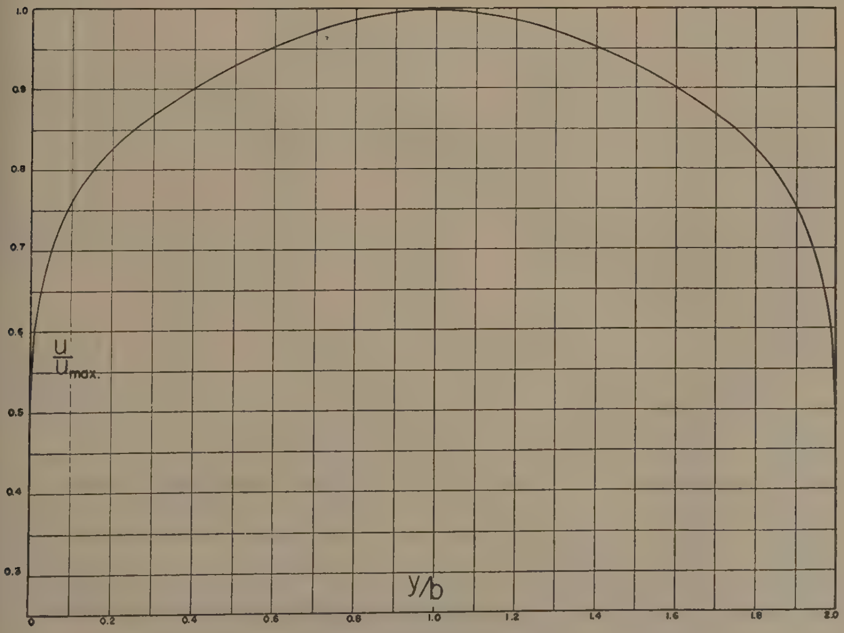

FIG. 1 VELOCITY DISTRIBUTION BETWEEN PARALLEL PLATES

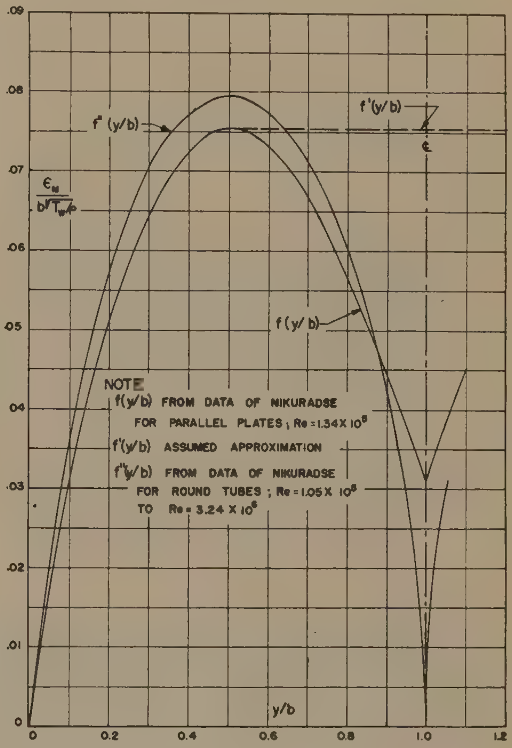

FIG. 2 EDDY DIFFUSIVITY OF MOMENTUM

Results of these calculations are shown in Table 1 and in Figs. 3, 4, and 5.

In order to explore the effect of the shape of the eddy-diffusivity curve near the center line of the channel, calculations were repeated after making the assumption that the eddy diffusivity remains constant at its maximum from $0.5 \leq y/b \leq 1.5$. This is shown as $f'(y/b)$ in Fig. 2. A paper by Brooks and Berggren (3) presents evidence in support of this assumption. There is some indication that eddy diffusivity goes to a maximum at the center line of a channel (4). One case was repeated using the eddy-diffusivity data which Nikuradse (5) obtained in circular tubes. For this case, ϵ_M was assumed to go to zero at the center line, as in Martinelli's paper. This variation of

$$\frac{\epsilon_M}{b\sqrt{\tau_w/\rho}}$$

TABLE 1 NUSSELT NUMBERS CALCULATED FOR ASYMMETRICAL SYSTEM

N_{Re}	N_{Pr}	N_{Pe}	N_{Nu}	Basis
10^5	0.005	5×10^3	18.4	$f(y/b)$
			19.6	$f'(y/b)$
10^6	0.01	10^4	29.1	$f(y/b)$
			31.2	$f'(y/b)$
			24.4	$f''(y/b)$
10^6	0.05	5×10^4	106	$f(y/b)$
			113	$f'(y/b)$
10^6	0.70	7×10^5	782	$f(y/b)$
			810	$f'(y/b)$
10^5	0.005	5×10^2	8.0	$f(y/b)$
			8.1	$f'(y/b)$
10^5	0.01	10^3	9.2	$f(y/b)$
			9.5	$f'(y/b)$
10^5	0.05	5×10^3	20.5	$f(y/b)$
			21.8	$f'(y/b)$
10^4	0.01	10^2	5.55	$f(y/b)$
			5.55	$f'(y/b)$

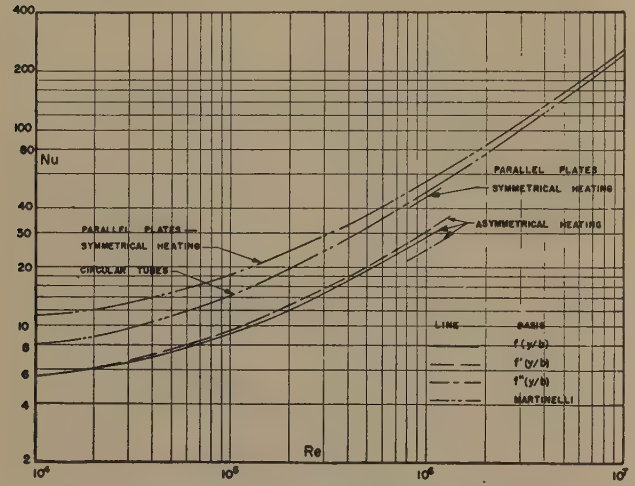

FIG. 4 COMPARISON OF CALCULATIONS FOR SYMMETRICAL AND ASYMMETRICAL CASES HAVING N_{Pr} = 0.01

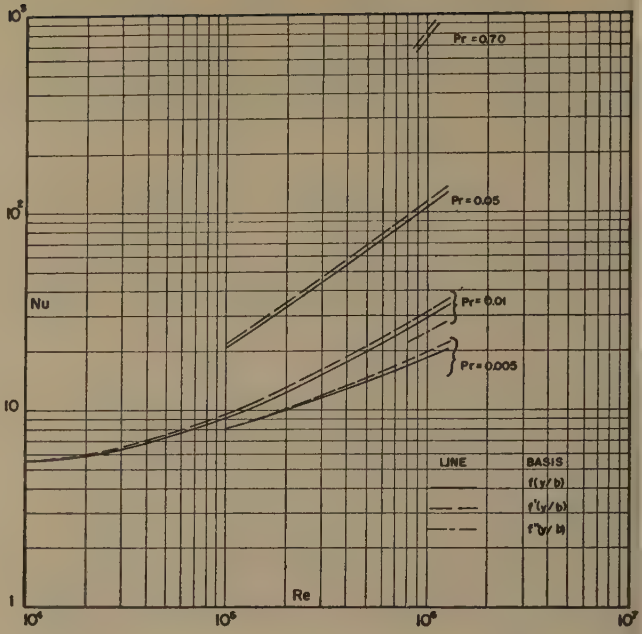

FIG. 3 PLOT OF NUSSELT NUMBERS CALCULATED FOR ASYMMETRICAL CASE

(Note that Prandtl, Nusselt, Reynolds, and Peclet numbers in Fig. 3 and subsequent figures correspond to text notation as follows: Pr = N_{Pr} in text; Re = N_{Re} in text; Nu = N_{Nu} in text; Pe = N_{Pe} in text.)

is shown in Fig. 2 as $f''(y/b)$. Nusselt numbers calculated by means of $f'(y/b)$ and $f''(y/b)$ are also shown in Table 1 and Figs. 3 and 4. The integrations involved in the calculations were performed graphically. In all cases, α was assumed to equal unity.

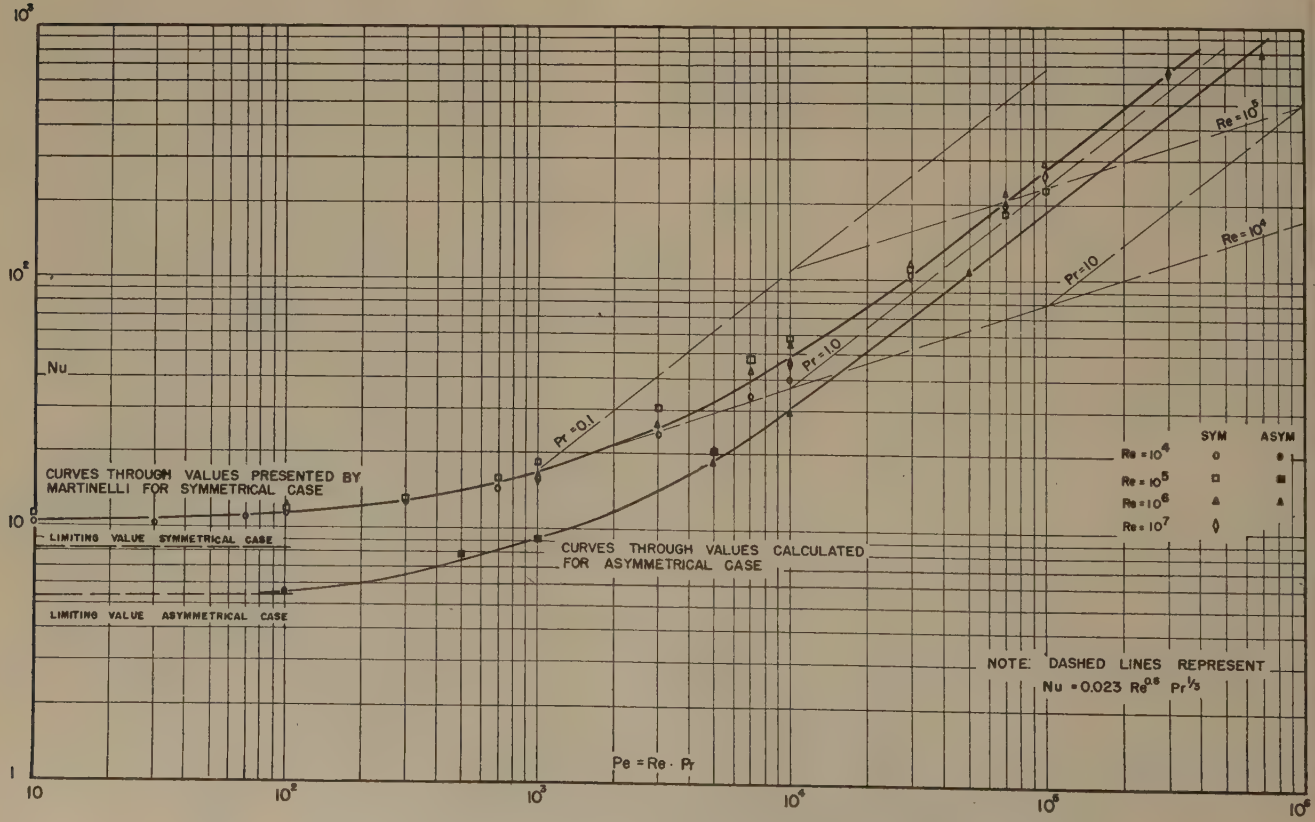

FIG. 5 N_{Nu} VERSUS N_{Pe} FOR SYMMETRICAL AND ASYMMETRICAL SYSTEMS

DISCUSSION OF FIGURES

As mentioned, Fig. 1 shows $u/u_{\max}$ versus y/b, and

$$\frac{\epsilon_M}{b\sqrt{\tau_w/\rho}} \text{ versus } y/b$$

is shown in Fig. 2. In Fig. 3 are plotted all the Nusselt numbers calculated for the asymmetrical case. It is hoped that the scope covered by these calculations can be broadened by additional calculations at a later date. Fig. 4 is a comparison of Nusselt numbers calculated for an asymmetrical flat-plate system with values presented by Martinelli for a symmetrical flat-plate system and circular tubes, Prandtl number being constant at 0.01 for all calculations shown. One value shown in Fig. 4 points out the difference in Nusselt number which results from using Nikuradse's experimental data for ϵ_M near the center line of the channel, rather than the method of Martinelli.

Fig. 5 utilizes a suggestion made by R. N. Lyon (6) for comparing the calculations for symmetrical and asymmetrical systems. It is seen that, when plotted as N_{Nu} versus N_{Pe}, a smooth curve fits all calculated points rather well. This method of presentation also points out under what conditions the equation,

$$N_{Nu} = 0.023\, N_{Re}^{0.8}\, N_{Pr}^{1/3} \ldots\ldots\ldots\ldots\ldots [18]$$

can be expected to hold, that is, if $N_{Pr} = 0.7$, Equation [18] and Martinelli's calculations are almost identical for all Peclet numbers greater than 2×10^4. Then, for symmetrical systems, it is indicated that Equation [18] may be used for cases having $N_{Pr} \geq 0.7$ and $N_{Pe} \geq 2 \times 10^4$, and Martinelli's calculations should be used for $N_{Pr} < 0.7$. For asymmetrical systems it is indicated that Equation [18] can be used for $N_{Pr} \geq 1.64$ and $N_{Pe} \geq 10^4$, and that the calculations furnished in this paper should be used for $N_{Pr} \leq 1.64$.

Limiting values of Nusselt number were taken from Norris and Streid (7) for the symmetrical case, and from Jakob (8) for the asymmetrical case. Both calculations are based upon constant heat flux and parabolic velocity distribution. One would expect the minimum for turbulent flow to be slightly higher since the velocity distribution is more nearly uniform than the parabolic distribution on which their calculations were based.

Figs. 6 and 7 are discussed in the following section of this paper.

RECOMMENDATIONS FOR AN EXPERIMENT

Fig. 6 shows a typical plot of

$$-\frac{1}{Q}\,\frac{dt}{d(y/b)} \text{ versus } y/b$$

for the turbulent core. From the development of the asymmetrical equation, it is seen that

$$-\frac{1}{Q}\,\frac{dt}{d(y/b)} = \frac{1 - y/2b}{f(y/b) + \dfrac{4}{\alpha N_{Pe}\sqrt{f/2}}}$$

in the turbulent region. In the symmetrical system

$$-\frac{1}{Q}\,\frac{dt}{d(y/b)} = \frac{1 - y/b}{f(y/b) + \dfrac{4}{\alpha N_{Pe}\sqrt{f/2}}}$$

The curves resulting from the use of $f'(y/b)$ and $f''(y/b)$ are also shown on Fig. 6. It should be possible to obtain the variation of eddy diffusivity of heat across a channel by making a traverse across the channel with two thermocouples a fixed distance apart. This should give direct experimental measurement of

$$-\frac{dt}{d(y/b)}$$

and, as seen in Fig. 6, the shape of the curve near the center line of the channel should indicate what happens to the eddy diffusivity. It is likely that velocity explorations in the same channel might yield more information on the evaluation of eddy diffusivity of momentum and, in turn, α.

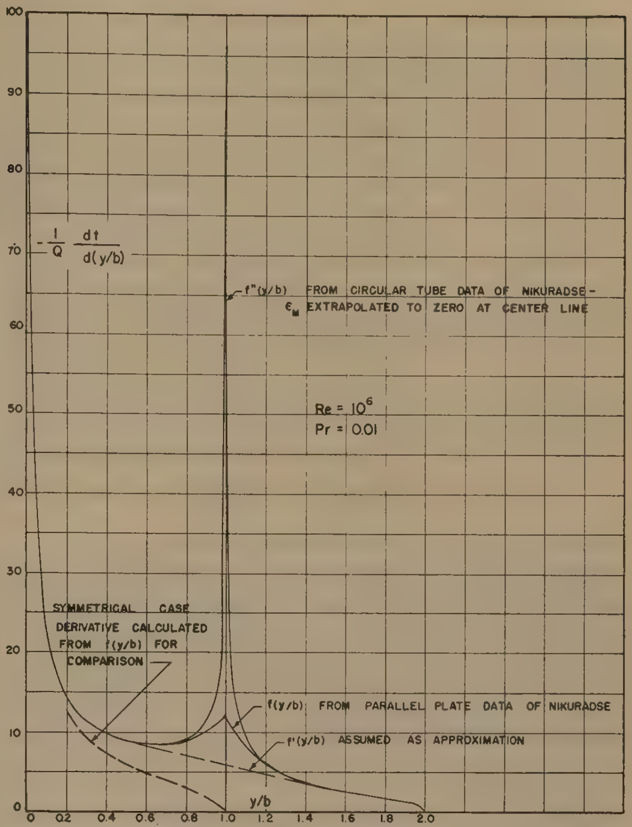

FIG. 6 PLOT OF CALCULATED TEMPERATURE DERIVATIVES SHOWING EFFECT OF EDDY-DIFFUSIVITY VALUES FOR AN ASYMMETRICAL CASE

Fig. 7 shows a typical plot of temperature distribution

$$\frac{t_w - t_3}{t_w - t_w'} \text{ versus } y/b$$

It is seen that the shape of this curve is also greatly affected by the shape of the eddy-diffusivity curve. This fact also may be utilized in studying the eddy diffusivity. The effects produced by $f(y/b)$, $f'(y/b)$ and $f''(y/b)$ are more pronounced in Fig. 6 than in Fig. 7, but careful experiments should enable one to explore the center region adequately, utilizing either of these types of plots as a basis for study.

A method for examining the asymmetrical heat-transfer coefficients would be in a "figure of eight" system in which the heat exchanger is composed of two rectangular channels having a common wall. Other confining surfaces of each stream should be insulated so that all heat is transferred through the wall between the two streams. By choosing the fluid and operating conditions so that the temperature difference between the two streams is not great enough to make the physical properties of the streams appreciably different, the heat-transfer coefficients should be the same on both sides of the wall, lending simplicity to an experimental study by eliminating the need to measure the temperature of the common wall.

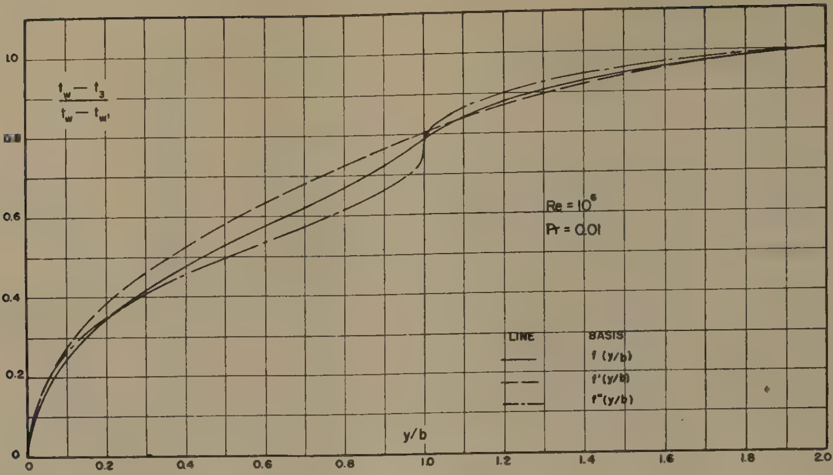

FIG. 7 PLOT OF TEMPERATURE DISTRIBUTION SHOWING EFFECT OF EDDY-DIFFUSIVITY VALUES FOR AN ASYMMETRICAL CASE

CONCLUSIONS

Theoretical work by Martinelli on heat transfer to molten metals has been extended to cover the asymmetrical system having fluid moving between two plane parallel walls, only one of which permits heat transfer to the stream. At low Peclet numbers, Nusselt numbers calculated in the manner outlined are roughly 50 per cent of the values presented by Martinelli for symmetrical systems. At high Peclet numbers, asymmetrical-system calculations are about 67 per cent of the symmetrical system calculations.

Suggestions have been made regarding an experimental approach for finding the variation of eddy diffusivity across a flat channel especially in the vicinity of the center line, and also for determining heat-transfer coefficients in asymmetrical systems.

This paper is not considered to be complete in itself, but only a brief extension of the work presented earlier by Dr. R. C. Martinelli.

ACKNOWLEDGMENT

The authors wish to express their thanks to T. W. Mullikin[4] for his assistance in making the calculations for this paper, and to R. N. Lyon[4] for making many helpful suggestions. The work was sponsored by the Atomic Energy Commission under Contract W-7405-ENG-26 with the Oak Ridge National Laboratory.

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Discussion

R. N. LYON.[5] It is of interest to compare the results of these calculations with those of Martinelli in an effort to find some roughly accurate general relationship between them. If the values of N_{Nu} for symmetrically heated parallel plates, for asymmetrically heated parallel plates, and for symmetrically heated tubes are plotted versus $N_{Pr} \times N_{Re}$ or Peclet number N_{Pe}, three lines can be drawn which approximate the three conditions.

Inspection of the three lines reveals the interesting fact that the following relations hold with reasonable accuracy up to N_{Pr} about 0.5:

$N_{Nu(\mathrm{sym})} = 3 + N_{Nu(\mathrm{CT})}$ at the same value of N_{Pe}

$N_{Nu(\mathrm{asym})} = 0.7\ N_{Nu(\mathrm{CT})}$ at the same value of N_{Pe}

$N_{Nu(\mathrm{asym})} = 0.7\ [N_{Nu(\mathrm{sym})} - 3]$ or $[0.7\ N_{Nu(\mathrm{sym})} - 2.1]$ at the same value of N_{Pe}

In all these cases D in $N_{Nu} = hD/k$ and $N_{Pe} = cu\gamma D/k$ is the usual $4A/P_w$ where A is the flow cross section in the channel and P_w is the wetted perimeter.

The theoretical significance of these approximations, while interesting for speculation, is probably not important. It is felt, however, that there may be a practical value in being able to convert from one situation to the other by such simple means.

R. C. MARTINELLI.[6] Since a paper by the writer[7] is used as a basis for the present paper, it should be pointed out that an error was made in that paper.[7] This error was first noted by Mr. R. N. Lyon.

The temperature distribution for the limiting case of $N_{Pr} = 0$ was shown in the former paper[7] as a series of broken straight lines. Actually the temperature distribution should have been parabolic. The equations in the reference are correct and give this parabolic curve, but a numerical error was made in establishing the curves, which gave the erroneous temperature distribution for $N_{Pr} = 0$. Luckily, this error seriously affects only calculations for $N_{Pr} <$ 0.003, and fluids with N_{Pr} 0.003 are rare. In order to set the record straight, however, the corrected values of $(t_w - t_m)/$

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[4] Oak Ridge National Laboratory, Oak Ridge, Tenn.

[5] Oak Ridge National Laboratory, Oak Ridge, Tenn.

[6] Associate Professor of Mechanical Engineering, University of California, Berkeley, Calif. Mem. ASME. Deceased January 9, 1949.

[7] See authors' Bibliography (1).

$(t_w - t_c)$ and the Nusselt modulus are tabulated herewith (Tables 2–5). In these calculations the friction factors for smooth pipes presented by Moody[8] were utilized.

TABLE 2 RATIO $(t_w - t_m/t_w - t_c)$ FOR CIRCULAR PIPES

N_{Re} / N_{Pr}	2000	10^4	10^5	10^6	10^7
0	0.600	0.564	0.558	0.553	0.550
0.0001	0.610	0.568	0.560	0.565	0.617
0.001	0.616	0.570	0.572	0.627	0.728
0.01	0.621	0.589	0.639	0.738	0.813
0.10	0.669	0.692	0.761	0.823	0.864
1.00	0.806	0.865	0.877	0.897	0.912
10	0.943	0.958	0.962	0.963	0.966
100	0.991	0.992	0.993	0.993	0.994

TABLE 3 RATIO $(t_w - t_m/t_w - t_c)$ FOR FLAT PLATES

$(N_{Re})_{FP}$ / N_{Pr}	4.89×10^3	2.20×10^4	2.16×10^5	2.12×10^6	2.10×10^7
0	0.745	0.718	0.711	0.707	0.704
0.0001	0.752	0.720	0.711	0.713	0.746
0.001	0.754	0.726	0.719	0.754	0.818
0.01	0.758	0.731	0.769	0.824	0.874
0.1	0.791	0.798	0.839	0.881	0.908
1.0	0.864	0.913	0.918	0.931	0.940
10	0.966	0.972	0.974	0.976	0.977
100	9.995	0.996	0.996	0.996	0.996

TABLE 4 NUSSELT MODULUS FOR CIRCULAR TUBES

N_{Re} / N_{Pr}	2000	10^4	10^5	10^6	10^7
0	5.90	7.02	7.17	7.23	7.27
0.0001	5.82	6.98	7.29	8.11	12.5
0.001	5.78	7.08	8.06	13.3	43.3
0.003	5.85	7.15	9.18	22.9	94.7
0.007	5.93	7.60	12.5	38.2	184
0.010	6.01	7.97	14.5	48.7	244
0.030	6.34	9.66	25.7	108	605
0.070	6.75	12.2	42.6	205	1224
0.100	7.10	14.0	53.0	271	1649
0.300	8.87	21.6	107	623	3993
0.700	11.3	31.2	179	1128	7666
1.000	12.9	36.7	217	1416	9953

TABLE 5 NUSSELT MODULUS FOR FLAT PLATES

$(N_{Re})_{FP}$ / N_{Pr}	2000	10^4	10^5	10^6	10^7
0	8.2	10.2	11.2	11.3	11.3
0.0001	8.3	10.2	11.2	12.1	16.5
0.001	8.3	10.2	12.0	16.2	46.0
0.003	8.4	10.3	13.1	26.6	104
0.007	8.5	11.0	16.1	43.2	196
0.010	8.6	11.2	18.3	54.0	260
0.030	8.7	12.6	30.5	116	640
0.070	8.9	14.8	48.0	220	1300
0.100	9.2	16.0	59.0	290	1730
0.300	10.2	23.8	110	658	4200
0.700	12.1	34.0	183	1160	7800
1.000	13.9	40.0	225	1450	10200

It is evident that the values of N_{Nu} are for all practical purposes the same as those shown in the writer's paper,[7] but the values of $(t_w - t_m)/(t_w - t_c)$ differ at the lower values of N_{Pr}, where the curves in the former paper were shown dotted. None of the conclusions reached in that paper is changed by this correction.

The extension by the authors of the analogy between heat and momentum transfer to asymmetrically heated channels is very interesting and provides a badly needed basis for analyzing heat transfer in asymmetrically heated annuli.

It is evident that the greatest difference between heat transfer in symmetrically and unsymmetrically heated channels occurs with molten metals. Experimental data for heat transfer to molten metals are necessary in order to verify the analysis presented.

The data of Corcoran, Roudebush, and Sage,[9] and Woertz and Sherwood,[10] may help clarify the question of the eddy diffusivity at the center of a channel.

AUTHORS' CLOSURE

The authors wish to acknowledge with appreciation the interest shown in this paper and the discussions by Dr. Lyon and Dr. Martinelli. It is pleasing to note that the relations presented in this paper have been applied by Dr. Lyon to the analysis of experimental data on sodium-potassium alloy flowing in an annulus.[11] The data appear to confirm the predictions within the limits of experimental error and the uncertainty regarding physical properties of the alloy. Dr. Lyon has recommended approximate equations which should be very useful to workers in this field.

Further research is being conducted at the University of Tennessee on heat transfer to mercury flowing in an asymmetrically heated rectangular channel, and it is hoped that the data will be available in the near future.[12]

It is with some shock that the authors must record their inability fully to acknowledge the contributions of Dr. Martinelli. He was entitled to say, "Non omnis moriar."

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[8] "Friction Factors for Pipe Flow," by L. F. Moody, Trans. ASME, vol. 66, 1944, pp. 671–684.

[9] "Temperature Gradients in Turbulent Gas Streams," by W. H. Corcoran, N. Roudebush, and B. H. Sage, *Chemical Engineering Progress*, vol. 43, 1947, p. 135.

[10] "The Role of Eddy-Diffusion in Mass Transfer Between Phases," by B. B. Woertz and K. T. Sherwood, Trans. AIChE, vol. 35, 1939, p. 517.

[11] "Heat Transfer at High Fluxes in Confined Spaces," by R. N. Lyon, PhD Thesis, University of Michigan, 1949.

[12] Department of Chemical Engineering, University of Tennessee, Knoxville, Tenn.

Heat Transfer to Water at High Flux Densities With and Without Surface Boiling

By FRANK KREITH[1] AND MARTIN SUMMERFIELD,[2] PASADENA, CALIF.

Surface coefficients of heat transfer have been obtained at rates of heat flux up to 3 Btu/(sq in.)(sec) for water flowing in stainless-steel tubes at mass-flow rates up to 5.4 lb per sec-sq in. The nonboiling forced-convection data were correlated by the Colburn equation $Nu_f = 0.023\ Re_f^{0.8}\ Pr_f^{1/3}$ within ± 5 per cent, but, with surface boiling, considerably larger *Nu* moduli were observed. When surface boiling occurred, the wall temperature was determined primarily by the pressure, being relatively insensitive to variations in velocity, heat flux, and bulk temperature. The maximum heat flux that could be removed under given conditions was limited by an unsteady-flow phenomenon. Data for pressure drop with and without nonboiling heat transfer are presented and correlated by an empirical viscosity correction factor.

NOMENCLATURE

The following nomenclature is used in this paper:

A = area
C = constant
c_p = specific heat
C_F = friction coefficient
$C'_F = \dfrac{C_{F\ (\text{isothermal})}}{(\mu_F/\mu_W)^{0.13}}$
E = voltage drop
f_c = heat-transfer coefficient
g = gravitational constant
J = conversion constant
j = dimensionless heat-transfer modulus $\left(\dfrac{Nu}{Re\ Pr}\right) Pr^{2/3}\left(\dfrac{\mu_W}{\mu_F}\right)^{0.14}$
j_f = dimensionless heat-transfer modulus $\left(\dfrac{Nu_f}{Re_f Pr_f^{1/3}}\right)$
ΔH = frictional pressure drop in head of liquid
K = thermal conductivity
L = length
$m = \dfrac{E^2}{2JK_1R_1}$
P = pressure
q = rate of heat flow
r = radius
R = resistivity
t = temperature
T_X = excess temperature
V = velocity
w = weight rate of flow
x = distance
Δx = tube-wall thickness
Re = Reynolds number
Nu = Nusselt number
Pr = Prandtl number
α = temperature coefficient of electrical resistivity
β = temperature coefficient of thermal conductivity
μ = absolute viscosity
ν = kinematic viscosity
ρ = mass density

Subscripts:

B = boiling
CS = cross section
f = arithmetic-mean temperature conditions
F = fluid bulk
FC = nonboiling forced convection
H = heat transfer
L = liquid
o = outside
W = wall
1 = at 0 F

INTRODUCTION

Uncertainities existing in means for predicting heat-transfer coefficients of liquids at severe temperature gradients adjacent to the surface, especially when the surface temperature exceeds the boiling temperature of the liquid, led to an experimental study of heat transfer at heat-flux densities above 0.5 Btu/(sq in.)(sec). The need for such heat-transfer data arises particularly in the design of liquid-cooled rocket motors where heat fluxes as high as 6 Btu/(sq in.)(sec) have been encountered, and in more recent power-plant developments.

EQUIPMENT

A choice among steam, gas combustion, or electric heating was considered as means of providing high heat fluxes, and the use of an electrical heating source was decided upon for the following reasons:

1 The heat-flux density is uniform.

2 Heat-flux densities up to 10 Btu/(sq in.) (sec) can be obtained.

3 The control of heat flux is relatively simple.

4 Many geometric configurations of cooling channels can be studied with the same over-all setup.

5 A reliable check on the thermal-energy output can be obtained by comparison with the electric-power input.

The following is a description of the equipment designed for the purpose of investigating heat transfer from a metal wall to a coolant at severe temperature gradients, for which the available information is inconclusive (1).[3] A line diagram illustrating the over-all setup of the equipment is shown in Fig. 1. The power is supplied by a 100-kva 440-volt step-down transformer whose secondary voltage can be varied from 20 to 40 volts in fifty-six steps by means of knife switches tapped from the primary wind-

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NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society. Paper No. 48—A-38.

[3] Numbers in parentheses refer to the Bibliography at the end of the paper.

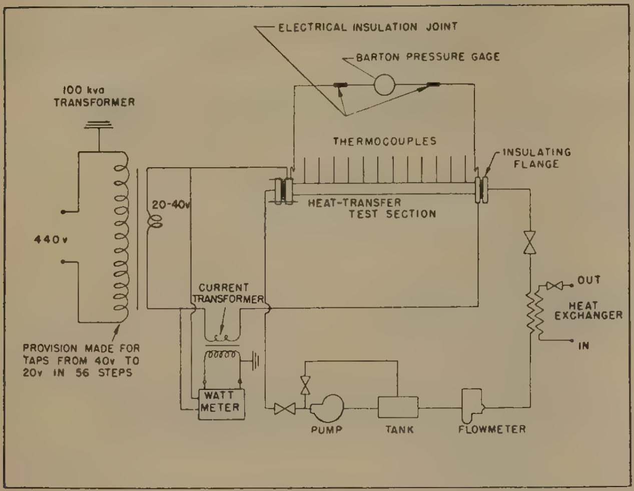

FIG. 1 FLOW AND WIRING DIAGRAM FOR ELECTRICALLY HEATED TEST SECTION

ing. The electric power is dissipated in the test section which acts as a resistance element and is cooled by the test liquid in forced circulation. The heat-flux output for a given resistance of the test section can be decreased to about one quarter of the maximum by changing the voltage potential. The maximum heat-flux density of a test section is only a function of the total resistance and the heat-transfer area is shown by the following equations

$$\frac{q}{A_H} = \frac{E^2}{R'A_H} \times \frac{1}{1.054} \quad [1]$$

$$R' = \frac{RL}{A_{CS}} \quad [2]$$

In practice heat-flux densities as high as 10 Btu/(sq in.)(sec) can be obtained by a proper balance between the cross-sectional area and heat-transfer area of the test section.

The test fluid was circulated by means of a Byron Jackson Hydroplex 10-stage centrifugal pump with a capacity of 50 gpm at 500 psi outlet pressure. The heated fluid was cooled by a three-pass tubular heat exchanger before returning to the tank. Flow measurements were taken in the return line by means of rotameters at flow rates below 1.5 lb per sec and by means of a double-orifice flowmeter at higher rates of flow. For the boiling heat-transfer tests a Grove model 161, constant-flow control valve was installed in the line between the pump and the test section to maintain a constant flow rate at different back pressures.

All test sections which were used in the experiments consist of stainless-steel type 304, commercially drawn tubes, whose ends are silver-soldered to a copper block as shown in Fig. 2. The copper blocks on each end are connected to the secondary outlets of the transformer. The inlet and outlet lines connecting the test section to the rest of the hydraulic flow system are insulated electrically from the test section with Durablo washers, Fig. 2. The physical dimensions of the test sections are given in Table 1, and Fig. 3 shows the over-all installation with a vertical test section.

To protect the test section and personnel, several safety switches were included in the electrical system. These switches were designed to shut off the power supply when the safety door to the knife switches was opened, when the pump was turned off, and when the temperature of the tube reached 1000 F.

EXPERIMENTAL TECHNIQUE

Measurement of Temperatures at Liquid-to-Wall Interface. The most important and also the most difficult task in heat-transfer studies is the determination of the temperature of the heat-transfer surface. It was not possible to measure a tempera-

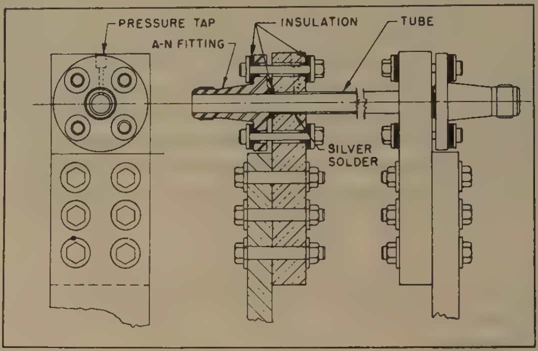

FIG. 2 ASSEMBLY DRAWING OF TYPICAL TEST SECTION

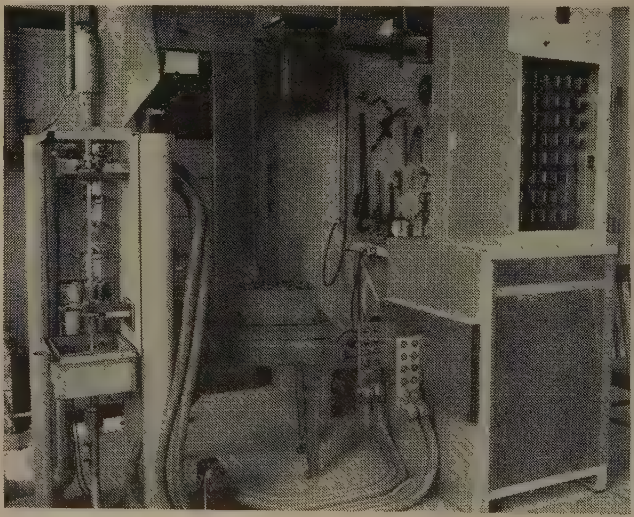

FIG. 3 VERTICAL TUBULAR TEST SECTION II AND TRANSFORMER SETUP

(*Upper left-hand corner:* Mixing chamber (insulation removed). *Left:* Test section with wall thermocouples. *Center:* Current leads and power panel. *Right:* Transformer and voltage switches.)

TABLE 1 PHYSICAL DIMENSIONS OF TEST SECTIONS[a]

	Friction test section	Heat-transfer test section I	Heat-transfer test section II	Heat-transfer test section III
Heated length in.	38.1	38.1	17.5	17.5
Inside diameter (ID), in.	0.527	0.535	0.587	0.587
Wall thickness, in.	0.049 ± 0.0015	0.0475 ± 0.0003	0.0208 ± 0.0003	0.0208 ± 0.0003
L/ID ratio	72.5	71.5	30	30
Surface rms (microinches)	50 to 60	50 to 60	45 to 55	40 to 45
Position of axis	Horizontal	Horizontal	Vertical	Vertical
Distance between pressure taps in.	39.625	...	...	...
Flow area, sq in.	0.218	0.225	0.2705	0.2705
Heat-transfer area sq in.	...	64.0	32.3	32.3

[a] All test sections were of stainless-steel type 304.

ture at the inner wall of the electrically heated tube directly. Therefore the temperature of the liquid-to-wall interface for the electrically heated test section was calculated from measurements of the outer-wall temperature, the voltage drop, and the electrical and physical properties of the test section. A spot-welding technique which is described elsewhere (2) was modified to install iron and constantan wires 0.005 in. OD to the outer surface of the tube.

Fig. 4 is a diagram of the spot welder which was developed at this laboratory for fusing small wires to a metal surface without any visible effects on the wire or the surface to which the wire is attached. The wire was held lightly to the surface by means of a pair of copper tweezers, and the condensers were discharged simultaneously. The resulting fusion weld was found to be strong in tension but weak in bending. Usually, however, not more than one out of twelve wires broke loose upon starting when the tube expanded suddenly and imposed a sudden bending moment on the wires. The wires of a thermocouple were installed about $^1/_{32}$ in. apart in a plane perpendicular to the axis of the tube.

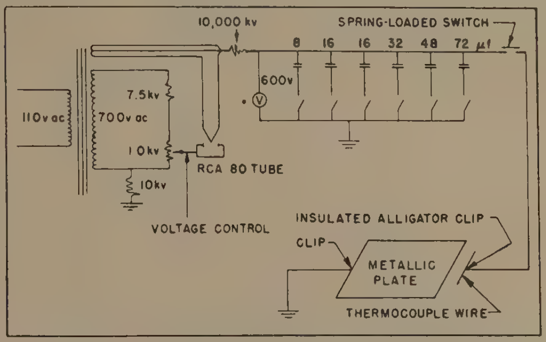

Fig. 4 Wiring Diagram of Spot Welder

The temperature of the inner-wall surface was calculated from Equation [3], expressing the temperature distribution in an electrically heated tube which is thermally insulated on the outer surface (reference 3 for derivation)

$$\frac{d^2t}{dr^2} + \left[\frac{1}{r} + \frac{K_1\beta}{K}\frac{(dt)}{(dr)}\right]\frac{dt}{dr} + \frac{E^2}{JKR} = 0 \ldots\ldots\ldots [3]$$

The equation was solved by the use of Taylor's series; the resulting expression for the temperature at the metal wall-to-liquid interface is

$$T_{W-L} = T_o - \frac{m\,\Delta x^2}{(1+\beta T_o)\,(1+\alpha T_o)}$$
$$- \frac{m\,\Delta x^3}{3r_o(1+\beta T_o)\,(1+\alpha T_o)} + \frac{m^2(3\alpha + 4\alpha\beta T_o + \beta)}{6\,(1+\beta T_o)^3\,(1+\alpha T_o)^3}$$
$$+ \frac{m}{4\,r_o^2\,(1+\beta T_o)\,(1+\alpha T_o)}\,\Delta x^4 + \ldots \ldots\ldots [4]$$

The location of the thermocouples for both test sections is shown in Figs. 5 and 6.

The electromotive forces of the thermocouples were recorded on a self-balancing Leeds and Northrup "Speedomax," using an ice bath as the cold junction. The possibility of interference with the operation of the Speedomax recorder by the alternating current present from the heater was investigated. The Speedomax is equipped with an internal filter, and the application of 60-mv (rms) across the thermocouple leads produced no observable effect regardless of phase with respect to the line voltage supplying the Speedomax. Presence of alternating-current voltages between the signal leads and ground caused serious trouble if the recorder case was grounded through an additional connection. If, however, no external ground was used with the Speedomax, no effect of alternating current was observed.

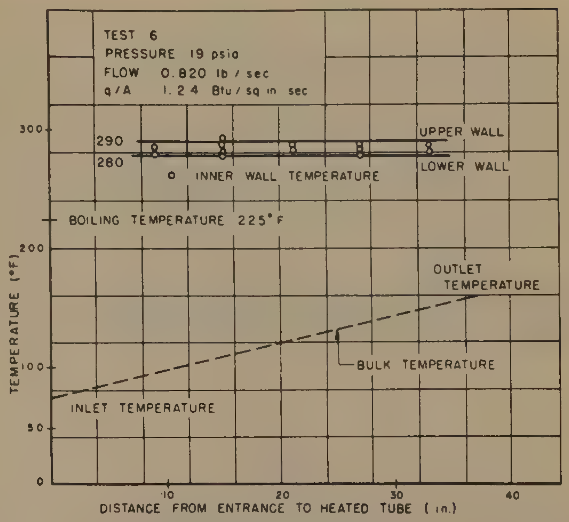

Fig. 5 Typical Test With Surface Boiling in Horizontal Test Section

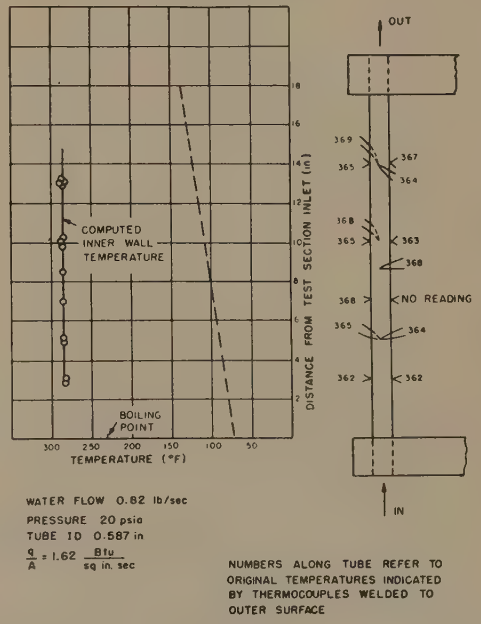

Fig. 6 Typical Test With Surface Boiling in Vertical Test Section

Measurement of Heat-Flux Density. One of the advantages of an electrically heated test section is the fact that the thermal-energy output can be checked accurately and independently by comparison with the electrical-power input.

The electrical-power input was measured with a Weston model 310 single-phase wattmeter which was connected directly across the voltage potential of the test section. The current was measured through a current transformer with a 500 to 1 transformation ratio.

The thermal-power output was determined from measurements of the rate of flow, average temperature of the liquid, and temperature rise of the liquid while it was flowing through the test section

$$q = w\,c_p\,(T_{\text{out}} - T_{\text{in}}) \ldots\ldots\ldots\ldots\ldots\ldots [5]$$

Entering-liquid temperatures were obtained by means of a precision thermometer and a thermocouple. The difference between inlet and outlet temperatures of the liquid was measured directly by using the inlet line as the cold junction and the outlet mixing chamber as the hot junction. Three iron-constantan thermocouples were connected in series to obtain a large electromotive force (emf) output and an average temperature. The differential temperature was measured on a Leeds and Northrup precision hand-balancing potentiometer several times during each test.

The heat-transfer area was clearly defined by the heated portion of the tube between the copper blocks. The heat-flux density could then be computed by either one of the following equations

$$\frac{q}{A} = \frac{w\,c_p\,(T_{\text{out}} - T_{\text{in}})}{A_H} \quad \ldots\ldots\ldots\ldots\ldots [6]$$

$$\frac{q}{A} = \frac{\text{kw}}{1.054 \times A_H} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots [7]$$

Measurements of Flow Rate and Pressure. Two rotameters in parallel were used to measure the flow rate of liquid up to 1.5 lb per sec, and a double-orifice metering section was used at higher rates of flow. The differential pressures across the orifice as well as the test section were measured by means of Barton pressure gages. A mercury manometer was used for calibration of the Barton gages. The absolute pressure of the liquid leaving the test section was recorded by three Bourdon-type pressure gages of 0 to 60, 0 to 200, and 0 to 400 psig range. The flowmeters were calibrated after installation gravimetrically and the test pressure gages were checked with a dead-weight tester. The accuracy of the flow and pressure measurements was well within 1 per cent.

Operating Procedure. The pump was turned on and the valves were set at the desired flow rate and test-section pressure. The Speedomax was turned on next, and the records of the thermocouple electromotive forces were checked against a precision mercury thermometer in the inlet line. The cooling water to the heat exchanger was turned on, and the voltage set at the desired value. After the Barton gages had been bled, the power switch was turned on. The mechanic and engineer took independent readings of Barton gages, pressure gages, mercury thermometer in the inlet line, and calibrating manometer during a test. The temperature difference between the inlet and outlet lines was measured by means of a hand-balancing Leeds and Northrup precision potentiometer and compared with the automatic record on the Speedomax tape. Steady state was assumed to have been reached when at least three successive cycles of records on the Speedomax were in agreement. The average time for one test was about 15 to 20 min.

After 60 to 70 min of operation, a check test was made in order to ascertain whether dirt had been deposited on the inner surface of the tube. These tests were made under conditions of flow, pressure, and power input as similar as possible to each other. If any deviations in the wall temperatures were noticed, the test section was cleaned with a dilute solution of hydrochloric acid. Dirt formation was noted in early tests with tap water, and in subsequent tests distilled water was used. All reported heat-transfer data were obtained with distilled water. Since the water was in equilibrium with atmospheric air at the suction side of the pump, it is probable that about 20 cc of air was dissolved per 1000 cc of water.

Accuracy of Results. The greatest possibility of error is involved in the determination of the temperature at the metal wall-to-liquid interface. One of the possible sources of error is the recording potentiometer whose accuracy is ±0.04 mv or 1.5 deg F. Another possible source of error is the nonuniformity of the tube. The tube thickness varied ±0.0003 in., and, since the computed temperature drop through the tube is a function of the square of the thickness (Equation [4]), the computed inner-wall temperature is accurate only to ±3 deg F under most severe conditions, i.e., maximum temperature drop through the metal wall.

The derivation of the equation for the temperature distribution in the electrically heated tube was based on certain assumptions which must be justified. The assumption of a zero temperature gradient at the outer surface, i.e., no heat loss to the surroundings, appears to be valid. Theoretical calculations based on natural convection data,[4] indicate that less than 0.1 per cent of the total energy was lost to the surroundings. These calculations are substantiated by extremely close checks between electric-power input and thermal-power output. The electrical resistivity of the stainless steel was obtained from reference (5), and information regarding its thermal conductivity was secured by private communication with the Bureau of Standards. The linear variation of these properties with temperature, which was assumed in the solution of Equation [3], agrees very well with the experimental data (5). Also, it should be pointed out that the results obtained with tubes of different length and thickness agreed within the accuracy of the measurements. Furthermore, the experimental results in the forced-convection region agree with literature data obtained under similar conditions with a steam-heated test section. The accuracy of the temperature at the inner-wall surface is considered to be within ±5 deg F in all tests.

The heat flux determined by Equation [6] agrees with the electric-power input within 1 per cent in all nonboiling tests. In this manner the validity of the mixed-fluid mean-temperature measurements could be checked. For the tests in the boiling region, the electric-power-input measurements were used for reducing the data because it was found that the temperature differences indicated by the thermocouples were consistently too low. It is believed that this discrepancy was caused by steam bubbles in the liquid stream whose latent heat had not been absorbed at the point of installation of the thermocouples near the outlet of the test section.

EXPERIMENTAL RESULTS

Pressure Drop in Turbulent Flow With Heat Transfer. It has been observed (1) that at a constant flow rate the friction drop decreases when heat is added to a liquid. In order to obtain a quantitative comparison of friction factors with and without heat transfer, a series of tests with water flowing in a stainless-steel tube, 0.527 in. ID and about 38 in. long, was performed at heat-flux densities from 0.5 to 1.6 Btu/(sq in.)(sec) in the Reynolds number range from 80,000 to 200,000, and the results were compared with those obtained without heating, Fig. 7. The Reynolds numbers for tests with heat transfer were computed at the bulk temperature halfway between the entrance and exit of the test section. Tap water was used for all friction tests.

It was found that the friction coefficient with heat transfer was from 5 to 16 per cent less than the isothermal value at the same bulk Reynolds number, and that the amount of reduction increased as the viscosity adjacent to the wall decreased. The ratios of the isothermal to nonisothermal friction coefficients are plotted versus the ratio of the viscosities at bulk temperature to that at the wall temperature in Fig. 8, in order to determine a correction factor. It was found that the nonisothermal friction coefficient was equal to the isothermal value at the same bulk Reynolds number times the ratio of viscosities at wall temperature to bulk temperature raised to the 0.13 power. The experimental friction factors are predicted within 3 per cent by the use of the correction factor for the Reynolds number range from

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[4] Reference (4), p. 243.

100,000 to 250,000 as shown in Table 2. These results agree substantially with conclusions presented by McAdams.[5]

Heat Transfer by Forced Convection Without Surface Boiling. The geometry of test section I was similar to that which was used by Hayes and Bartol (6) in a study of forced-convection heat transfer to water at Reynolds numbers and heat-flux densities comparable with the condition of the tests described in the present paper. It was expected that if the data were reliable, the tests in which no boiling occurred would show good agreement with the literature results.

Fig. 9 provides a comparison of the nonboiling data obtained in the electrically heated tube with the results from reference (6), which were obtained in a steam-heated test section; the widely used Colburn equation is also shown on the same graph (7)

$$j_f = \frac{Nu_f}{Re_f Pr_f} Pr_f^{2/3} = 0.023\, Re_f^{0.2} \ldots\ldots\ldots\ldots [8]$$

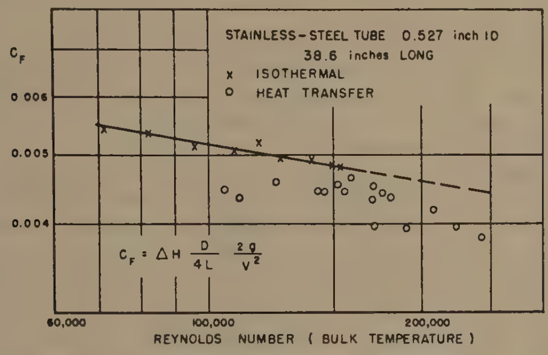

FIG. 7 FRICTION FACTOR VERSUS REYNOLDS NUMBER FOR HEAT TRANSFER AND ISOTHERMAL FLOW WITHOUT SURFACE BOILING

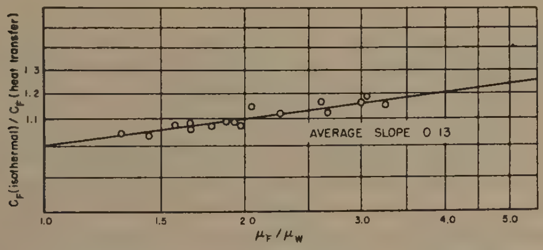

FIG. 8 CORRECTION FACTOR FOR FRICTION COEFFICIENT WITH HEAT TRANSFER WITHOUT SURFACE BOILING

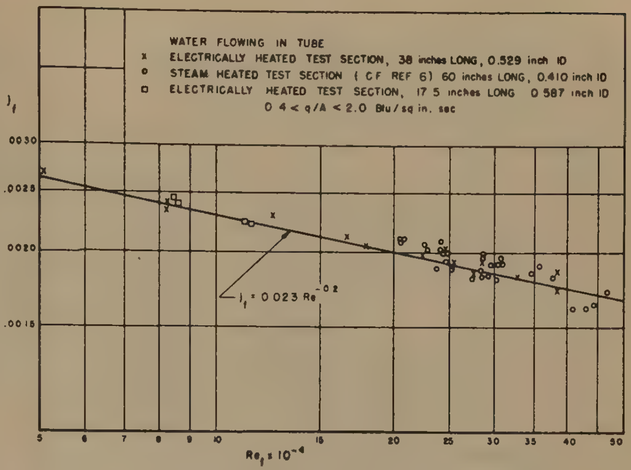

FIG. 9 j_f-FACTOR VERSUS REYNOLDS NUMBER FOR FORCED CONVECTION WITHOUT SURFACE BOILING

The agreement between the nonboiling data obtained at this laboratory and those reported in the literature is seen to be excellent in the experimental Reynolds number range from 50,000 to 400,000, and at heat-flux densities from 0.5 to 1.7 Btu/(sq in.)(sec). The results obtained with test section II are also shown on the same graph, and the agreement is equally good.

It should be pointed out, however, that the change of viscosity in the laminar boundary layer for water is relatively small (never larger than 2), and the effect of a severe temperature gradient in more viscous liquids such as aniline may cause deviations from Equation [8]. In future tests it is planned to extend the experiments to more viscous liquids.

The values of convective-film conductances, used for the evaluation of the Nusselt modulus, represent point values. Since the heat output is constant over the entire length of the tube, the heat-transfer coefficient could be evaluated at any point along the tube. In order to eliminate entrance effects, all test results were evaluated at distances larger than 12 diam from the entrance. The change in bulk temperature was assumed to be linear along the tube. The results of one typical forced-convection test are shown in Fig. 10, and the complete data and calculations for the other tests can be found in reference (3).

For engineering calculations, it is often more convenient to use a Reynolds number evaluated at the bulk temperature. Sieder and Tate (8) have correlated heat-transfer data using the bulk temperature to evaluate the *Re*, *Nu*, and *Pr* moduli and correcting for the temperature gradient by means of the

TABLE 2 FRICTION IN TUBE WITH HEAT TRANSFER[a]

$$C_F = \frac{\Delta H}{V^2} \cdot \frac{D}{2Lg}$$

Test no.	w lb per sec	T_F deg F	T_W deg F	$Re_F \times 10^{-3}$	ΔP psi	$C_F \times 10^3$	$\mu_F \times 10^5$ (lb-sec/sq ft)	$\mu_W \times 10^5$ (lb-sec/sq ft)	μ_F/μ_W	$(\mu_F/\mu_W)^{0.13}$	$C_{F\text{iso}} \times 10^3$	$C_F' \times 10^3$	$C_{F\text{exp}}/C_F'$
1	3.04	90	126	171	9.56	4.52	1.600	1.115	1.48	1.048	4.72	4.50	1.004
2	3.04	92	138	175	9.33	4.41	1.565	1.00	1.57	1.060	4.70	4.43	0.998
3	3.03	95	159	180	9.06	4.39	1.515	0.845	1.79	1.079	4.68	4.34	1.021
4	2.18	115	154	158	5.01	4.64	1.235	0.875	1.41	1.046	4.79	4.58	1.015
5	2.17	111	170	151	4.88	4.56	1.285	0.780	1.65	1.067	4.82	4.52	1.008
6	2.16	114	202	155	4.74	4.47	1.245	0.630	1.98	1.093	4.80	4.39	1.025
7	1.84	92	173	105	3.42	4.48	1.565	0.765	2.05	1.098	5.13	4.67	0.960
8	1.82	97	219	110	3.22	4.33	1.480	0.570	2.60	1.132	5.10	4.51	0.973
9	3.06	90	144	170	8.96	4.51	1.615	0.950	1.70	1.071	4.73	4.42	0.964
10	3.06	90	181	172	8.48	3.99	1.600	0.620	2.58	1.131	4.72	4.17	0.957
11	3.06	99	248	190	8.34	3.93	1.445	0.480	3.01	1.154	4.63	4.01	0.987
12	3.06	89	157	170	9.17	4.32	1.615	0.855	1.89	1.086	4.73	4.36	0.998
13	2.15	92	163	124	4.88	4.60	1.566	0.818	1.91	1.088	4.98	4.58	1.007
14	2.15	106	240	143	4.70	4.43	1.350	0.505	2.67	1.136	4.87	4.30	1.045
15	2.15	107	283	146	4.45	4.23	1.335	0.410	3.26	1.166	4.85	4.16	1.018
16	3.90	99	246	241	13.2	3.82	1.450	0.485	2.98	1.152	4.45	3.86	0.990
17	3.90	92	187	224	13.75	3.98	1.566	0.690	2.27	1.112	4.50	4.05	0.988
18	3.90	86	136	209	14.5	4.20	1.676	1.013	1.65	1.067	4.57	4.28	0.984

[a] Commercial stainless-steel, 5/8-in. drawn tube; fluid, water; tube ID, 0.527 in.; tube length, 39.625 in.

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[5] Reference (4), p. 121.

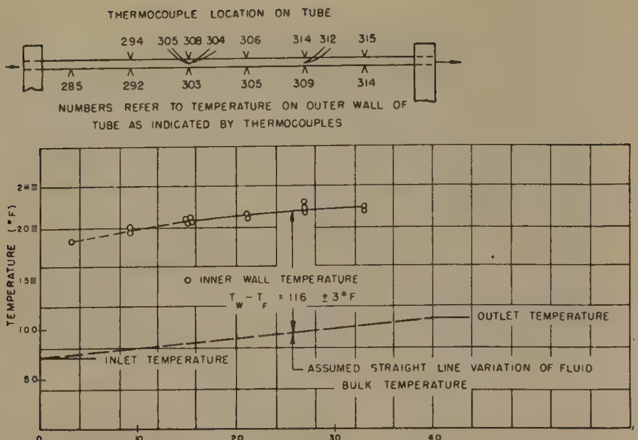

Fig. 10 Typical Test Without Surface Boiling in Horizontal Test Section

viscosity ratio of wall to bulk raised to the 0.14 power

$$j = \frac{Nu}{Re\,Pr} Pr^{2/3} \left(\frac{\mu_W}{\mu_F}\right)^{0.14} = 0.027\,Re^{0.2} \ldots\ldots\ldots [9]$$

The experimental results agree with Equation [9] within ±8 per cent in the range of Reynolds numbers, based upon the bulk temperature, from 25,000 to 300,000.

Attempts to correlate the experimental results with theoretical approaches such as presented by Boelter, Martinelli, and Jonassen (9) and Mattioli (10) were postponed until data on more viscous liquids are available. A cursory check on the y^+ value for test 1, Table 3, gave a value slightly below 5, which is in agreement with reference (9).

Forced Convection With Surface Temperature Above Boiling Temperature of Liquid. There is a considerable amount of information available for forced-convection heat transfer to a one-phase system (liquid or gas), but the forced-convection field, where the temperature of the surface in contact with the liquid is above the boiling temperature of the liquid and local boiling occurs, has not been studied to the same extent (11, 12, 13, 14). In particular, there is little information available in regard to the influences of pressure, velocity, shape of flow conduit, and surface finish on heat transfer with surface boiling to subcooled liquids at high velocities.

In a series of preliminary tests with distilled water in test section I, the effect of surface boiling in the liquid adjacent to the wall upon heat transfer was investigated. Some of the results are illustrated in Fig. 5, where the bulk temperature and metal-surface temperature for a typical test are plotted as a function of distance from the entrance to the heated tube. It can be seen that the wall temperature is constant over the entire length, regardless of the liquid bulk temperature which rises from 68 to 160 F. The lower portion of the tube is at a temperature of about 10 deg F below that of the upper portion. It is believed that this temperature stratification is due to an effect of gravity. The difference between the wall temperature and the boiling temperature of the liquid is constant over the entire test section.

The data for test section I, presented in Table 3, are only a fraction of the total obtained. The results of a larger number of tests were discarded because later check tests indicated that a deposit had formed on the inner wall, especially in that portion of the tube where the water was the hottest. All data for test section I, which are presented in Table 3, with the exception of test 5, were obtained within 50 min after the tube was cleaned.

Based upon the experience obtained in these preliminary experiments, test section II was constructed, and subsequently, data in the pressure range from 20 to 200 psia, the velocity range from 6 to 13 fps, and heat-flux range from 0.7 to 3.0 Btu/(sq in.)(sec) were obtained. The bulk temperature and metal-surface temperature for a typical test are plotted as a function of distance from the entrance to the heated tube, Fig. 6. Test section II was in a vertical position, Fig. 2, and it should be noted that there is little difference in wall temperature at any cross section. An additional test section, identical with test section II in physical dimensions but smoother in surface finish, was constructed and used in four tests. The data obtained in tests with sections II and III are summarized in Table 4.

The results of thirty-eight tests at mass-flow rates held within the range 0.72 to 0.82 lb per sec are shown in Fig. 11. The excess temperature T_X, defined as the difference between the wall temperature and the boiling temperature, is shown as a function of test pressure for heat fluxes from 0.7 to 2.8 Btu/(sq in.)(sec). The pressure drop in the test section was always less than 0.25 psi, and the test pressure was assumed constant from entrance to exit. The excess temperature was found to decrease with pressure at a given heat flux following approximately the relationship

$$T_X = C(P)^{-3/4} \ldots\ldots\ldots\ldots\ldots\ldots [10]$$

This agrees quantitatively with experimental results in nucleate pool boiling of saturated liquids where it was observed (15, 16, 17, 18, 19, 20, 21) that the temperature difference required to remove a given amount of heat decreases with an increase of the pressure on the system. Fig. 12, which was obtained by cross-plotting the average lines from Fig. 11, shows the excess temperature necessary to remove a given heat flux up to 3 Btu/(sq in.)(sec) at pressures of 20, 30, 50, and 200 psia for the flow-rate range mentioned previously.

In order to determine the influence of velocity upon the heat transfer, a series of tests at a mass-flow rate of 1.45 lb per sec was made, and the results are plotted in Fig. 13 as q/A versus T_X, where, for comparison, the results from Fig. 11 are cross-plotted for the same pressures. Experimental data at 25, 63, 103, and

TABLE 3 HEAT TRANSFER WITH SURFACE BOILING IN HORIZONTAL TEST SECTION I

Test No.	w lb per sec	P psia	T_B deg F	q/A Btu/(sq in.)(sec)	T_F deg F	T_W deg F	$T_W - T_B$ deg F	$Re_F \times 10^{-3}$	T_{out} deg F
1A	0.853	18	222.5	1.22	111	284	61.5	60	168
1B	0.853	18	222.5	1.22	156	283	60.5	90	
2A	1.07	21	228	1.60	110	291	60.5	74	168
2B	1.07	21	228	1.60	155	294	63.5	110	
3	1.05	45	274.5	1.60	112	314	39.5	74	162
4	0.553	52	284	1.57	192	334	50.0	73	243
5A	0.597	59	291.5	1.57	133	338	49.5	51	233
5B[a]	0.597	59	291.5	1.57	185	347	55.5	76	
6A	0.820	19	225	1.24	127	286	61	66	170
6B	0.820	19	225	1.24	157	285	60	84	
7A	0.332	16	216	0.42	95	248	32	19.6	144
7B	0.332	16	216	0.42	134	246	30	28.8	
8A	1.05	150	358.5	1.60	115	357	...	75.6	73

[a] Some surface scale found after this test.

TABLE 4 HEAT TRANSFER WITH SURFACE BOILING IN VERTICAL TEST SECTIONS II AND III

Test no.	w lb per sec	P psia	T_F deg F	$T_{w\,avg}$ deg F	T_{in} deg F	q/A Btu/(sq in.)(sec)	T_{out} deg F	T_X deg F
				Test Section II				
1	0.805	157	362	NB[a]	80	0.73	109	...
2	0.807	158	362.5	NB	80	1.10	124	...
3	0.801	158.5	363	377.5	80	1.58	144	14.5
4	0.809	158	362.5	383	80	2.10	164	20.5
5	0.810	110	334.8	360.8	81	2.10	165	26
6	0.810	110	334.8	355.3	81	1.59	144	20.5
7	0.809	110	334.8	347.8	81	1.12	126	13
8	0.810	110	334.8	NB	81	0.73	110	...
9	0.810	65.5	298.5	NB	82	0.73	111	...
10	0.815	65.5	298.5	321.5	82	1.12	126	23
11	0.775	60.5	293.2	326.2	82	1.58	148	33
12	0.788	61	293.8	336.3	82	2.15	170	42.5
13	0.790	36	261	316.5	83	2.15	171	55.5
14	0.785	36	261	302.5	83	1.62	150	41.5
15	0.785	36	261	297	83	1.13	129	36
16	0.784	36	261	279.5	83	0.73	113	18.5
17	0.820	22.5	234.2	264.2	84	0.73	113	30
18	0.822	22.5	234.2	281.2	84	1.13	128	47
19	0.820	22.5	234.2	291.2	74	1.62	138	57
20	1.444	103	330	NB	76	1.13	101	...
21	1.440	103	330	NB	76	1.62	112	...
22	1.440	103	330	351	76	2.10	123	21
23	1.438	104	330.7	366	76	2.76	138	35.3
24	1.440	168	367.5	379	76	2.73	137	11.5
25	1.470	25	240	260.7	75	1.13	100	20.7
26	1.437	24.6	239	281	77	1.62	113	42
27	1.432	24.6	239	289	77	2.14	125	50
28	1.428	24.6	239	294	77	2.78	140	55
29	1.448	63	296	342	77	2.78	139	46
30	1.440	63	296	333	77	2.12	125	37
31	1.438	63	296	315	77	1.62	113	21
32	0.725	17	219.5	274.5	76	1.14	127	55
33	0.768	63	296	319	79	1.13	127	23
34	0.738	40.5	268	299.5	80	1.13	129	31.5
35	0.722	27.5	245.5	285.5	80	1.13	131	40
36	0.720	16	216	271	80	1.15	132	55.5
37	0.770	202	382.5	401.5	81	2.08	168	19
38	0.765	168	367.5	388.5	81	2.08	169	21
39	0.768	102.5	329.5	360	81	2.10	169	30.5
40	0.768	62.5	295.5	334.5	81	2.13	171	39
41	0.730	40.5	268	320	81	2.14	176	52
42	0.780	16	216	276	81	2.15	170	60
43	0.795	193	379	404	82	2.76	194	25
44	0.790	139	352.5	384	82	2.77	195	31.5
45	0.785	78	310	354	82	2.78	196	45
46	0.765	45.5	275	327	82	2.80	200	52
47[b]	0.802	25			82	2.78		
				Test Section III				
1	0.720	64	297	317	81	1.64	155	20
2	0.726	37.5	263	293	81	1.63	154	30
3	0.726	29.5	249	287	81	1.63	154	38
4	0.726	153.5	360	371	81	1.65	154	11

[a] NB refers to forced convection, no boiling.
[b] Test section melted.

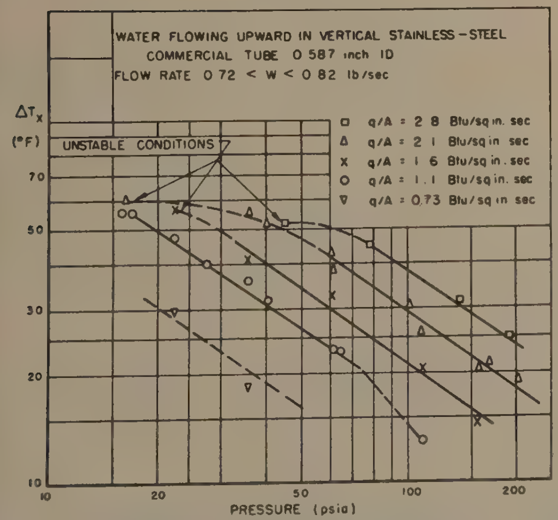

FIG. 11 EXCESS TEMPERATURE VERSUS PRESSURE FOR WATER IN VERTICAL TUBE WITH SURFACE BOILING

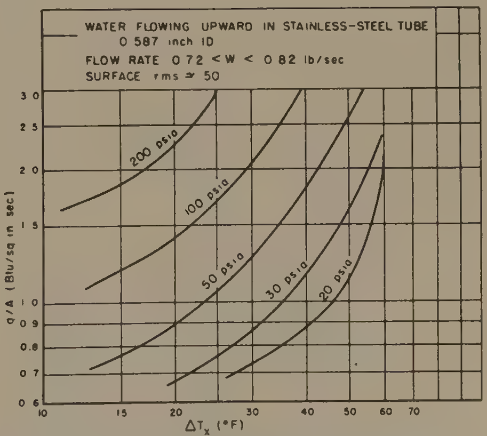

FIG. 12 HEAT FLUX VERSUS EXCESS TEMPERATURE FOR WATER IN VERTICAL TUBE WITH SURFACE BOILING

168 psia were obtained at heat-flux densities up to 3 Btu/(sq in.)(sec). The influence of velocity upon the temperature excess is seen to become smaller as the heat flux increases.

Fig. 14 presents a comparison between forced convection and boiling heat transfer. A bulk temperature of 150 F and mass-flow rates of 0.75 and 1.45 lb per sec were assumed in the construction of the graph, which shows the temperature difference between wall and liquid necessary to remove a given rate of heat per unit area for forced convection with and without surface boiling. It can be seen that when the surface temperature reaches the boiling point of the liquid, the mechanism of heat transfer changes, and large increases in heat flux are accompanied by only minor increases in wall temperature. It can be seen that the influence of velocity is small compared with that of liquid pressure.[6]

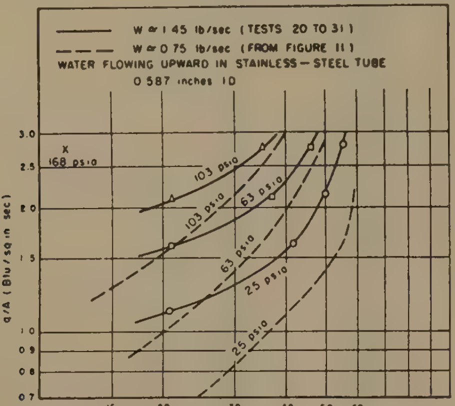

FIG. 13 INFLUENCE OF VELOCITY UPON EXCESS TEMPERATURE FOR WATER IN VERTICAL TUBE WITH SURFACE BOILING

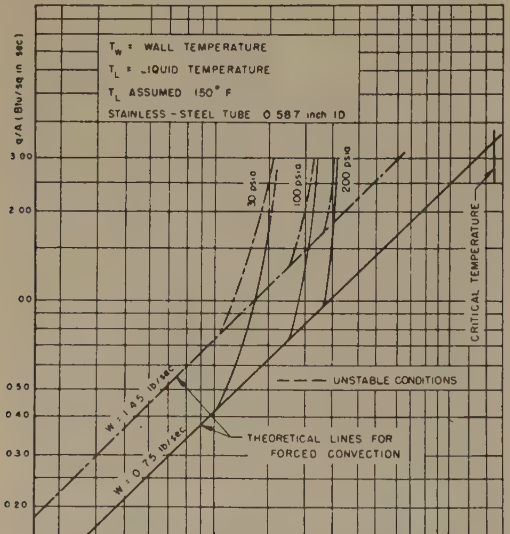

FIG. 14 COMPARISON OF FORCED-CONVECTION HEAT TRANSFER WITH AND WITHOUT SURFACE BOILING

Accepting the conventional definition for the film conductan (Equation [11]), the ratio of conductances with and witho surface boiling can be estimated readily from Fig. 14 using th following relation

$$f_e = \frac{q}{A(T_{\text{wall}} - T_{\text{fluid}})} \quad \ldots\ldots\ldots\ldots\ldots [1$$

$$\frac{f_{e\,\text{boil}}}{f_{e\,\text{no boil}}} = \frac{\Delta T_{\text{no boil}}}{\Delta T_{\text{boil}}} \quad \ldots\ldots\ldots\ldots\ldots [12$$

where ΔT refers to the temperature difference between wall an bulk of fluid. It should be borne in mind, however, that th relationship holds only for the bulk temperature specified. F any other bulk temperature, similar curves can be constructe using the results presented in Fig. 13.

Other investigators have shown that the nature of the soli surface has an important effect on heat transfer in the regio of surface boiling (21, 22, 23, 24, 25). To examine this effect, tes section II was polished with emery cloth to a 40-microinch roughness, and a series of tests at q/A = 1.6 Btu/(sq in.)(sec) wa made. The measurements are plotted in Fig. 15, together wit data derived from Fig. 11, which were obtained with the sam test section in the unpolished condition (50-microinch roughness It can be seen that the excess temperature for the polished su face is about 25 per cent less than that for the original surface at the same heat flux and test pressure. Although this result i rather unexpected, it agrees with theoretical considerations an experimental results by Larson (25), which indicate that th wettability of the surface and not the surface finish influences th excess temperature. Larson also observed an increase of T when the surface of a stainless-steel ebullator was roughened.

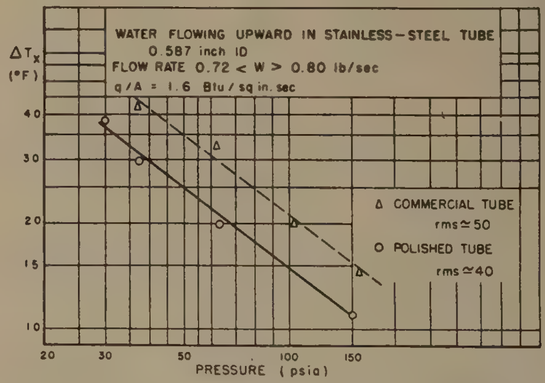

FIG. 15 INFLUENCE OF SURFACE FINISH UPON EXCESS TEMPERATUR

Not enough tests were made to establish this effect over the ful range of heat flux, velocity, and pressure. It is believed that th uncertainty in wall temperature associated with this effect is no appreciable in the design of cooled rocket motors or other hea exchangers.

The foregoing discussion applies only to stable heat-transfe and flow conditions. No definite criterion for stability ca be presented at this time. However, the following remarks ar presented to describe the region where operation ceased to b stable:

When the conditions of a test were such that, because of a lo velocity and high heat flux, the excess temperature exceeded 5 deg F and the bulk temperature near the outlet approached th boiling point within 60 to 80 deg F, the test section began to vibrate and the differential pressure oscillated considerably.

After this unstable condition had been reached, a further decrease in velocity resulted in a failure of the test section near th outlet end. The tube melted suddenly over a relatively narrow circumferential strip about $^1/_8$ in. in width as shown in Fig. 16. A safety thermocouple, designed to cut off the power supply when the outer-tube temperature reached 1000 F, failed to operate even though its response time was less than 2 sec, and its location less than $^1/_4$ in. from the melting zone.

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[6] Experimental results substantially in agreement with those reported here have been obtained by Minden and Kennel at the Massachusetts Institute of Technology under the direction of W. H. McAdams, working with an annular test section, the inner surface of which was heated. (Private communication from M.I.T. dated January 21, 1948.)

It is not clear whether the excess temperature difference or the bulk temperature (or a combination of both) limits the amount of heat which can be removed at a given velocity. An attempt was made to investigate the bulk-temperature effect by recirculating the water without cooling in the heat exchanger. Unfortunately, the results of this test could not be interpreted because the heated water caused corrosion in the mild-steel pump, and an iron-oxide was precipitated on the wall of the test section. It is necessary either to install a high-quality filter on the downstream side of the test section, or to heat the water after it leaves the pump prior to entering the test section.

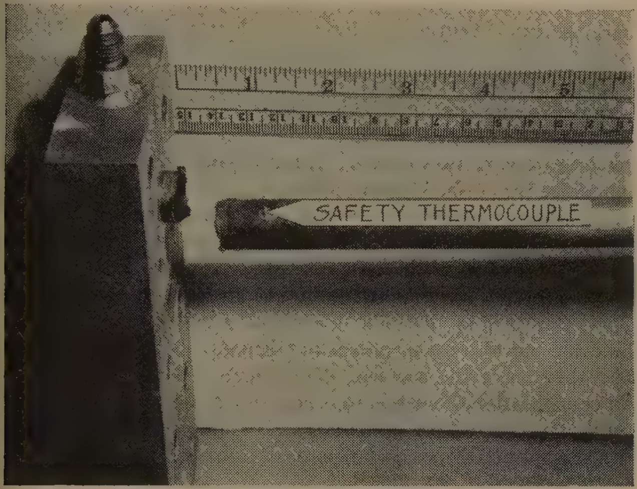

FIG. 16 FAILURE OF TEST SECTION II

These limits to the surface-boiling regime of heat transfer will be considered further. At this time, however, it is important to note that such limits exist, and to avoid them in the design of cooling systems.

CONCLUSIONS

The significant conclusions reached in the present experimental study of heat transfer at high heat-flux densities to water flowing in a tube are as follows:

1 When heat is transferred to water in turbulent flow without surface boiling in a tube, the friction coefficient at equal Reynolds numbers decreases in accordance with the following ratio

$$\frac{C_F(\text{heat transfer})}{C_F(\text{isothermal})} = \left(\frac{\mu_W}{\mu_L}\right)^{0.13} \qquad [13]$$

2 The conventional Colburn equation

$$Nu_f = 0.023\, Re_f^{0.8}\, Pr_f^{1/3}$$

correlates experimental data for water flowing in a tube in the Reynolds number range from 50,000 to 350,000 and at heat-flux densities from 0.5 to 2.0 Btu/(sq in.)(sec) within 5 per cent.

3 The heat-transfer coefficient between a metal wall and a liquid in turbulent flow cannot be predicted from the Colburn equation when the surface temperature exceeds the boiling point of the liquid.

4 The surface temperature necessary to transfer a given amount of heat from a wall to a liquid when surface boiling occurs is primarily a function of the pressure of the liquid. At a specified flow velocity, the surface temperature can be reduced below that required for nonboiling convective heat transfer by reducing the pressure to permit surface boiling.

5 The temperature difference between the surface and the boiling point of the liquid seems to be the driving potential in heat transfer by forced convection with surface boiling. This temperature difference for water at a given heat flux was found to decrease at a velocity of about 7 fps as the reciprocal of the absolute pressure raised to the $^3/_4$ power. This agrees qualitatively with results obtained in pool boiling.

6 The amount of heat which can be absorbed by the liquid at a given pressure and velocity was limited by an unstable flow phenomenon similar to that which occurs in flow with severe cavitation. No definite criterion for the limit of operational stability can be presented at this time.

The research is being continued to investigate the influence of high heat-flux densities on the behavior of other liquids such as methyl alcohol, aniline, and oils. The influence of curvature of the flow channel will be examined, in both the boiling and nonboiling regimes. Such effects as the operational instability found at high rates of heat flux and the change in heat-transfer characteristics with surface condition will be explored more fully. Information on these questions should prove useful in the design of cooling systems for rocket motors and other high-heat-flux devices.

ACKNOWLEDGMENTS

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Discussion

R. F. Larson.[7] It should be borne in mind, and perhaps emphasized, that these tests are primarily tests in heating liquid, water, making use of the intermediate processes of boiling and subsequent condensation. Obviously, the heat-transfer rate q/A will be a function of the extent to which this entire process is completed, either within the test section or beyond it. With this in mind it is clear that the results are also functions of the geometry of the apparatus, the diameter and length of the tube, as well as its position. Obviously, an inclined or horizontal tube would not give the same results as a vertical one, thus the rates obtained should not be taken as representative of rates obtainable from other possible arrangements.

The entire process is complex and cannot be simplified readily to considerations of film coefficients alone, unless the complete process is completed at the heat-transfer surface or adjacent to it. The authors, perhaps wisely, did not calculate and correlate such film coefficients for the case of intermediate boiling. Failure to recognize this may account for lack of correlation of coefficients possible between the results of various investigators. The unstable conditions are undoubtedly dynamic phenomena set up by vapor-locking of the tubes at the top and a passage over to the film-boiling range, with resultant overheating and failure. The entire process is then not completed within the tube, and the safe capacity of the tube has been surpassed.

That a smaller temperature excess, $T_w - T_b$, above the normal boiling temperature follows upon polishing and perhaps otherwise cleaning the tube surface is indicated from both physical and surface-energy considerations. The condition for bubble initiation on the wall surface depends upon the wettability or the adhesion tension $(\sigma_{SV} - \sigma_{LS})$, or better yet, the combination $\sigma_{LV} + (\sigma_{SV} - \sigma_{LS})$, which establishes the liquid superheat necessary.[8] This property is not homogeneous along a metal surface, but varies according to homogeneity of crystalline structure, presence of impurities, and cleanliness.

A high adhesion tension means a high degree of wettability and this will determine the size and shape of bubbles formed on the surface, and how soon they will be released and swept away to be later condensed, if such occurs. The ebullition process is intermittent with large fluctuations of liquid superheat at a point, the result being a very efficient type of turbulence created just where it will do the most good, at the very surface film. A relatively strongly wetted surface such as stainless steel promotes this process, the nuclei being sufficient in number, yet not too many, and small enough to prevent film boiling from being established even at fairly high rates of heat transfer. Geometric consideration of surface roughness will explain why surface bubbles adhere longer on rough surfaces, and, hence, inhibit the desirable fluctuations and resulting surface turbulence. Surface deposits not only add resistance to heat transfer but may change the surface wettability considerably.

More experimentation of this type using other heating surfaces and other liquids, thus giving a wide range of wettability or surface free energy, are in order. That a lower pressure drop accompanies the surface-boiling phenomenon, in spite of the added turbulence, is probably due to the presence of the vapor phase and offers added motive for investigating further the use of boiling to improve over-all heat transfer. However, how far one can go in this direction without encountering vapor lock or vapor film boiling depends upon not only control of surface ebullition but also upon control and disposal of the vapor bubbles after they leave the surface. The entire process, both boiling and condensation, if such is to occur, should be considered in the design of such a heat exchanger. If boiling and evaporation are the aim, then disposal of vapor is of prime importance.

Authors' Closure

The authors appreciate Professor Larson's comments on this paper. They agree that more experimental work of this type with other heating surfaces and other liquids is in order. Since the publication of this paper, an experimental program with aniline and *n*-butyl alcohol has been completed, and the results of this study will be published soon.

As Professor Larson points out, the heat-transfer process with surface boiling is complex and does not lend itself readily to a theoretical treatment. However, the results from recent tests at this laboratory with an annular test section (with forced flow) and with a stainless-steel strip submersed in a subcooled water bath[9] have provided additional clues to the mechanism of surface boiling and its limitations. Data were obtained regarding the upper limit of heat flux which can be removed by water with surface boiling, and also the factors causing the breakdown of the heat-transfer processes were studied qualitatively. The annular test section which was used for these tests was formed between a stainless-steel heating tube ($^3/_8$ in OD) and a transparent pyrex jacket ($^5/_8$ in. ID). The heated section of the tube was 7 in. long. The test section was located in a vertical position, and the cooling water from the laboratory supply system entered at the bottom.

Burnout conditions were investigated at heat-flux values from 2 to 8 Btu/sq in. sec, in the velocity range from 2 to 14 fps, and the pressure range from 15 to 75 psia. The results of the tests are correlated in a manner suggested by McAdams in Fig. 17 of this closure, where the heat flux at burnout is plotted against the product of the entrance velocity times the subcooling tem-

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[7] Professor of Mechanical Engineering, University of Illinois, Urbana, Ill. Mem. ASME.

[8] Author's bibliography (25).

[9] "Photographic Study of Bubble Formation in Heat Transfer to Subcooled Water," by F. C. Gunther and F. Kreith, Heat Transfer and Fluid Mechanics Institute, ASME Publication, May, 1949, pp. 113–138.

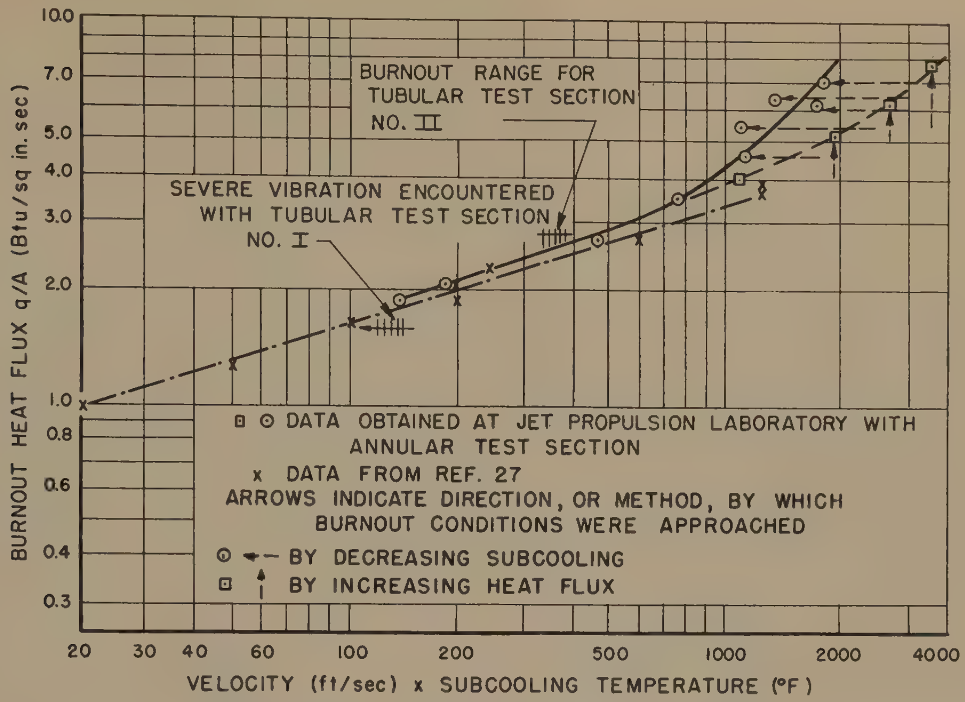

Fig. 17 Burnout Conditions for Annular Test Section
(Data for tap water.)

perature (which is defined as the temperature difference between saturation temperature at the test pressure and the bulk temperature of the liquid). The results obtained by McAdams, et al,[10] in burnout tests with water are superimposed on the graph of Figure 17. McAdams used an annular test section which was 3.75 in. long and was formed between a stainless-steel tube 0.25 in. OD and a jacket 0.77 in. ID. The following data are also shown in Fig. 17:

1 Burnout range which had been encountered with the tubular test section No. II.

2 Range of unsteady flow conditions (vibrations) with the tubular test section No. I.

It can be seen from an inspection of Fig. 17, that the results obtained with test sections of widely varying geometry are in agreement.

In passing, it is of interest to point out that the burnout heat flux appears to depend upon the suddenness with which the critical conditions are approached. Two methods of approaching burnout conditions were utilized during the tests for which the data are shown in Fig. 17:

1 In some tests, burnout conditions were approached at a constant heat flux (after steady-state cooling conditions had been established) by decreasing the amount of subcooling by reducing the pressure on the test section and/or by decreasing the velocity of the coolant liquid. (Solid curve.)

2 In an alternate approach to burnout conditions, the heat flux was increased in steps of about 0.1 Btu/sq in. sec, and the flow and pressure were maintained constant. However, for each heat-flux level, which was determined by the voltage across the heater tube, it was necessary to apply the electric power instantaneously to the test section. (Dashed curve.)

When technique (2) was used, burnouts occurred at higher velocities and at lower bulk temperatures (more subcooling) than when burnout conditions were approached by method (1). The directions or methods by which burnout conditions were approached are indicated in Fig. 17 by arrows. It is believed that when burnout conditions were approached by method (2), the rate of heating was so rapid that the critical range of surface temperature was traversed in a time which was comparable to or less than that required to establish equilibrium surface boiling conditions, and that the temperature of the tube could have exceeded momentarily the equilibrium temperature, resulting in a premature burnout. Based upon the photographic observations reported[9] previously, the time required to establish the surface-boiling heat-transfer mechanism is estimated to be from 0.001 to 0.005 sec, while the time available in approach (2) is estimated to be of the order of 0.002 sec at a heat-transfer rate of 4 Btu/sq in. sec, and becomes even less at larger values of heat flux.

A clue to the reason why the geometry of the test section is of relatively small importance upon the burnout limits was obtained from visual inspection and photographs of the boiling mechanism in the annular test section. It was observed that when the bulk temperature of the water in the test section was subcooled 75 deg F or more, only a very small amount of vapor existed in the main stream. Furthermore, the burnouts in the annular test section were not caused by a vapor lock at the top of the annulus, but rather by local vapor blanketing. It appears that this blanketing phenomenon is insensitive to the over-all geometry of the test section.

Additional information regarding the mechanism of surface boiling has been obtained from high-speed motion pictures of the bubble growth and collapse cycles on a flat strip heater.[9] The conclusions drawn from these data are substantially in agreement with the foregoing argument, namely, that the heat transfer process in the surface-boiling regime is dependent only on local conditions adjacent to the surface, and is relatively independent of the over-all geometry of the system.

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[10] "Heat Transfer at High Rates to Water With Surface Boiling," by W. H. McAdams, J. N. Addoms, and W. E. Kennel, Department of Chemical Engineering, Massachusetts Institute of Technology, December, 1948. (Report issued by Argonne National Laboratories.)

A Ventilated Thermal-Insulation Structure for High-Temperature Marine Power Plants

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This paper presents a method of design for a ventilated thermal-insulation structure suitable for high-temperature marine power plants. Such a structure is of substantially smaller thickness than the conventional type when designed for the same heat leak to the machinery space. This reduction, however, is obtained at the expense of increased heat leak from the thermodynamic working substance. Nevertheless, the cost in terms of plant fuel consumption may not be excessive. The ventilating air, which carries away the greater portion of the energy, is shown to have moderate requirements of flow rate, number of ducts, and blower power. The conclusions of the analysis are supported by some preliminary test results.

Nomenclature

The following nomenclature is used in the paper:

D_i = inside diameter of primary-insulation layer
D_o = outside diameter of primary-insulation layer
G = mass velocity of ventilating air, lb/(sec-ft^2)
h_f = fluid film conductance, Btu/(hr-ft^2-deg F)
k = thermal conductivity, Btu/(hr-ft^2-deg F/ft)
l = length of duct over which ventilating air passes, ft
q_h = heat leak from the working substance, Btu/(hr-ft^2 of hot area)
q_v = heat leak into the ventilating air, Btu/(hr-ft^2 of hot area)
q_∞ = heat leak into the machinery space, Btu/(hr-ft^2 of hot area)
R_f = thermal resistance of fluid film in ventilating space (hr-ft^2-deg F)/Btu
R_{f1}, R_{f2} refer to inner and outer surfaces
R_o = thermal resistance of thin outer blanket insulation, (hr-ft^2-deg F)/Btu
R_r = equivalent thermal resistance for radiation energy transfer (hr-ft^2-deg F)/Btu, across ventilating passage
R_w = thermal resistance of primary-wall insulation, (hr-ft^2-deg F)/Btu
T_c = temperature of outer surface of ventilated annulus, deg F
T_f = temperature of ventilating air, deg F
T_h = temperature of hot surface being insulated, deg F
T_o = temperature of outside surface of insulation, deg F
T_w = temperature of hot surface of ventilated space, deg F
T_∞ = temperature of machinery space, deg F
$\Delta T_h = T_h - T_\infty$
$\Delta T_f = T_f - T_\infty$
z = thickness of insulating material
δ = annulus width

Introduction

The orthodox design of thermal insulation for power-plant service is based primarily upon the objective of conserving the energy of the thermodynamic working substance. However, additional functions must be satisfied for many specialized installations. If a high-temperature plant must be installed in a limited space where operating personnel are in continuous attendance, as in a marine installation, a most important function is to reduce the heat leak to the machinery space. Where highly stressed components—structural support members, or pressure-vessel walls—come adjacent to high-temperature elements, an important function is to keep the stressed components in a region of low temperature, whereby less expensive alloys or smaller-size sections may be used.

In conventional insulation structures the heat leak from the working substance is of the same magnitude as the heat leak to the machinery space. Consequently, minimizing one will minimize the other, and two of the specified functions tend to lose their identities. In some of the high-temperature plants for merchant-marine or naval use it is essential, however, to maintain a distinction between these objectives. This will be demonstrated by the specific consideration of a high-efficiency marine gas-turbine plant relative to the conventional marine steam plant.

The conventional gas-turbine plant requires a rather elaborate cycle if it is to have a specific fuel consumption and flat economy characteristic comparable to the steam power plant. For instance, the plant described in a paper by C. R. Soderberg,[3] includes two stages of compression with intercooling, a high-effectiveness regenerator, and two turbines with reheat. This type of plant has a specific flow rate relative to that of a steam plant of 5:1. In the hot-duct portions of the cycle, the comparable working-substance density ratio, gas-turbine plant to steam plant, is about 1:6. In addition, because of the amplified cost of flow friction in gas-turbine plants, the allowable flow velocities are somewhat lower. For these three reasons hot-duct flow areas tend to be 40 to 50 times greater in the gas-turbine plant. The result is that hot-duct surface areas requiring insulation are on the order of 7 times greater for a given length of duct. Further, the temperature difference to be insulated against is perhaps 2 or 3 times that for a comparable steam plant.

It is evident, then, that the application of a conventional-type insulation to a gas-turbine plant, of a thickness suitable for a steam plant, will result in a machinery-space heat leak of 10 to 20 times that of the steam plant. Even with a greatly increased machinery-space ventilation, a suitable thickness of a conventional insulation structure will be on the order of 15 to 25 in. It is believed that the bulk and weight of insulation of this thickness will be a severe handicap in the adoption of a gas turbine as a marine prime mover.

Inasmuch as the gas turbine must have large flow passage areas to handle the volume flow rates, the use of conventional pressure vessels and ducting with externally applied insulation requires heavy sections of high-alloy material. The alternative

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society. Paper No. 48—A-46.

[3] "A Marine Gas Turbine Plant," by C. Richard Soderberg, Ronald B. Smith, and A. T. Scott, Trans. The Society of Naval Architects and Marine Engineers, vol. 53, 1945, pp. 249–289.

of placing the pressure-retaining walls behind the protection of a layer of thermal insulation in a region of lower temperature may make low-grade alloys and reduced weight possible, even though the total volume maintained under pressure is increased.

As an answer to this problem, it is the purpose of this paper to describe and predict performance for a ventilated type of thermal-insulation structure, which allows independent control over the three insulation functions; control of heat leak from the working substance, heat leak to the machinery space, and the temperature of stressed container walls. Such a structure, with an overall thickness of 4 to 5 in., for a hot-wall temperature of 1500 F, will reduce the heat leak to the machinery space to substantially less than that of the conventional insulation on a steam plant. Temperatures at any point through the insulation may be readily predicted for the purpose of securing safe conditions for stressed members which may be located there. The ventilation requirement with respect to blower and duct size and power requirements are shown not to be excessive, and the heat leak from the thermodynamic working substance is also shown to be of a tolerable magnitude. Some preliminary test results of such an insulation structure are presented to demonstrate the validity of the analysis.

Description of System

For brevity, only one form of ventilated thermal-insulation structure will be considered specifically. Many other variations are possible and may possess advantages.

The specific system to be considered in this paper is shown schematically in Fig. 1. It consists of a layer of primary insulation several inches in thickness adjacent to the hot surface. This layer controls the magnitude of the heat leak from the working substance, q_h. Next to this layer is a ventilated annulu Outside of the ventilated annulus is a layer of blanket insulatio $^1/_4$ to $^1/_2$ in. thick, which with the coolant flow serves to minimiz the heat leak to the machinery space, $q_\infty = q_h - q_v$, to the desired magnitude. This insures a colder outer surface which reduces direct radiation from the structure to operating personnel, a factor of considerable importance from the point of view o comfort.

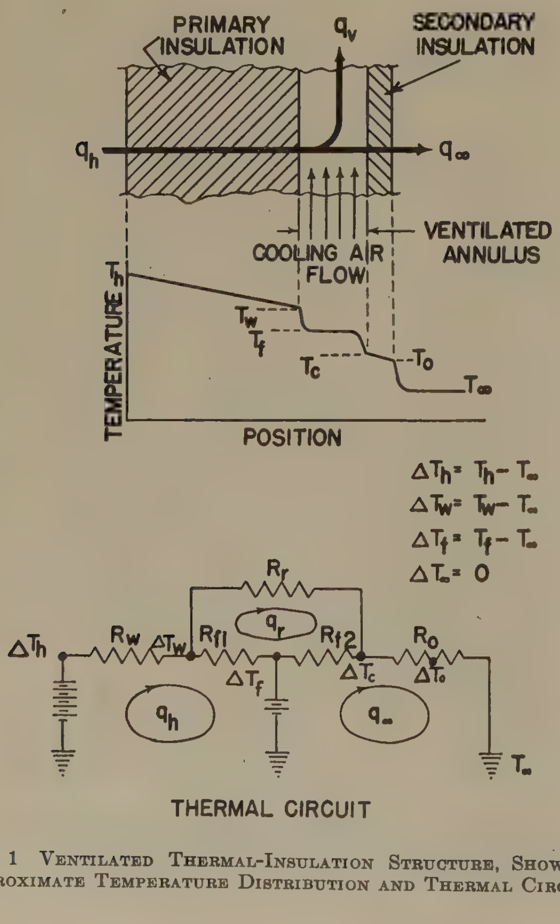

Fig. 1 Ventilated Thermal-Insulation Structure, Showing Approximate Temperature Distribution and Thermal Circuit

Also shown in Fig. 1, for steady-state conditions, is a description of the temperature distribution, and a "thermal-circuit" diagram exhibiting the relation of the component thermal resistances and thermal potentials (temperatures). In this circui all the temperature potentials are referred to the "ground temperature" or ambient temperature, T_∞, as a datum. Thus, fo instance, $\Delta T_h = T_h - T_\infty$ is the temperature of the hot surfac referred to the ambient temperature as the datum; R_w denote the thermal resistance of the primary insulation layer; R_{f1} an R_{f2} denote the thermal resistances for convection heat transfe from the inner wall of the ventilated annulus to the coolan and from the coolant to the outer wall, respectively; R_r denotes th effective resistance for radiant heat transfer from the hotter t colder walls of the ventilated annulus; R_0 denotes the series parallel combination resistance of the secondary-insulation laye and the combined radiation and convection from the outer surface to the machinery space.

The coolant temperature, T_f or ΔT_f, is increasing continuall along the flow path as a result of absorption of the q_v energy This factor determines the desirable coolant flow length and th frequency of the ventilation ducts in a given installation.

As can be seen from the foregoing description, the structur considered is the same in principle as the air-wall construction o modern naval boilers or the tail-pipe insulation of aircraft turbojet units.

Analysis

The thermal circuit shown in Fig. 1 suggests the metho of analysis. Three Kirchoff-law-type equations may be writte for the three circuits indicated. These may be solved simultaneously to yield the thermal currents q_h, q_∞ in terms of ΔT_h ΔT_f, and the component resistances. Then from energy-balanc consideration for steady-state conditions

$$q_v = q_h - q_\infty$$

Before proceeding with this analysis, a simplification will be accomplished by specifying that the convection resistances R_{f1} an R_{f2} are sensibly the same; thus

$$R_{f1} = R_{f2} = R_f$$

This is permissible in that, aside from the small differences i film viscosity, the flow conditions on the two walls of the ventilated space are closely alike.

The three circuit equations (rate equations for heat transfer are

$$\Delta T_h - R_w q_h - (q_h - q_r)R_f - \Delta T_f = 0 \qquad [1$$

$$\Delta T_f - (q_\infty - q_r)R_f - R_0 q_\infty = 0 \qquad [2$$

$$R_r q_r + (q_r - q_\infty)R_f + (q_r - q_h)R_f = 0 \qquad [3$$

From Equations [1], [2], and [3], respectively

$$q_h = \frac{\Delta T_h - \Delta T_f}{R_w + R_f} + \frac{R_f}{R_w + R_f}\, q_r \qquad [1a$$

$$q_\infty = \frac{\Delta T_f}{R_f + R_o} + \frac{R_f}{R_f + R_o}\, q_r \qquad [2a$$

$$q_r = \frac{R_f}{R_r + 2R_f}(q_\infty + q_h) \ldots\ldots\ldots\ldots [3a]$$

Substitution of Equations [1*a*] and [2*a*] into [3*a*] yields

$$q_r = \left(\frac{\Delta T_h - \Delta T_f}{R_w + R_f} + \frac{\Delta T_f}{R_f + R_o}\right) \Bigg/ \left(\frac{R_r + 2R_f}{R_f} - \frac{R_f}{R_w + R_f} - \frac{R_f}{R_f + R_o}\right) \ldots\ldots [4]$$

This result introduced into Equations [1*a*] and [2*a*] provide the desired relations

$$_h = \left(\frac{\Delta T_h - \Delta T_f}{R_w + R_f}\right) + \left(\frac{1}{R_w + R_f}\right) \left. \frac{\Delta T_h(R_f + R_o) + \Delta T_f(R_w - R_o)}{\left(\frac{R_w}{R_f} + 1\right)\left(\frac{R_o}{R_f} + 1\right)(R_r + 2R_f) - (R_f + R_o) - (R_w + R_f)} \right] \ldots\ldots\ldots\ldots [5]$$

$$_\infty = \left(\frac{\Delta T_f}{R_f + R_o}\right) + \left(\frac{1}{R_f + R_o}\right)\left[\begin{array}{l}\text{Same terms as used in}\\ \text{brackets, Equation [5]}\end{array}\right] \ldots [6]$$

As indicated previously, after evaluation of q_h and q_∞, the ventilating-air energy transfer may be determined readily as

$$q_v = q_h - q_\infty \ldots\ldots\ldots\ldots [7]$$

Inspection of the thermal circuit, Fig. 1, demonstrates that the hot-surface temperature of the ventilated space may be evaluated from

$$T_w = T_h - q_h R_w \ldots\ldots\ldots\ldots [8]$$

Similarly

$$T_e = T_\infty + q_\infty R_0 \ldots\ldots\ldots\ldots [9]$$

The outside-surface temperature likewise may be determined from a breakdown of the resistance R_o into its components of secondary-insulation layer resistance, and the combined radiation-convection resistance for the transfer to the machinery space from the outer surface.

Equations [5] to [9] lend themselves to ready tabular form calculations for any specified combination of ΔT_h, ΔT_f, R_w, R_f, R_o, R_r. As an illustration of the results of this type of calculation, Figs. 2 to 6 are presented.

The values of variables used for these calculations are as follows:

Hot-surface temperature, T_h = 1200 F
Machinery-space temperature, T_∞ = 90 F (Hence ΔT_h = 1110 F)
Outside resistance R_o = 0.45 + 0.55 = 1.00 (hr-deg F-ft^2 of hot wall)/Btu (The first term is for the secondary insulation layer and the other for the convection-radiation resistance from the outer surface to the machinery space, based upon data given by McAdams.[4])
Ventilation-space radiation resistance
  R_r = 0.56 (hr-deg F-ft^2 of hot wall)/Btu
  (Based upon radiation between parallel planes and a surface emissivity of 0.9)
Ventilation-space convection resistance R_f, as a function of mass velocity G, calculated from

$$h_f = \frac{1}{R_f} = 4.0\ G^{0.8} \text{ (McAdams}^4\text{)}$$

(Note that the flow Reynolds number for G = 0.5 lb/(sec-ft^2) is about 3000; thus the foregoing turbulent-flow equation should be applicable for $G > 0.5$ lb/[sec-ft^2].)

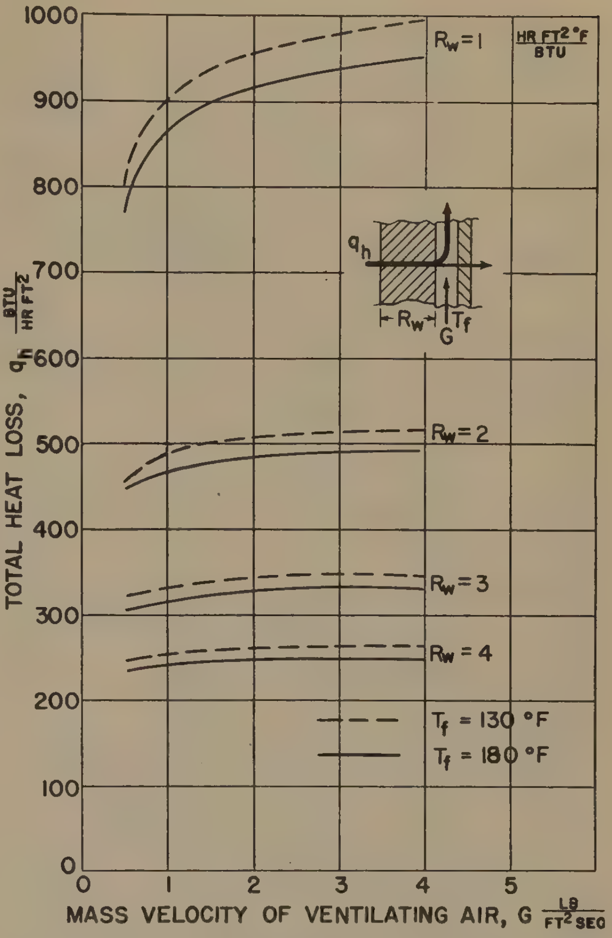

FIG. 2 TOTAL HEAT LEAK FROM WORKING SUBSTANCE, VARIATION WITH MASS VELOCITY OF VENTILATING AIR AND PRIMARY-INSULATION RESISTANCE

(Hot-surface temperature, 1200 F; machinery-space temperature, 90 F.)

It is assumed that this insulation is applied either to a flat surface or a relatively large cylindrical surface so that the heat-transfer area remains sensibly invariant with radial-flow position. Insulation thicknesses for cylindrical surfaces are necessarily greater than for plane surfaces.

The results presented in Figs. 2 to 6, will now be discussed. Fig. 2 presents the heat leak from the working substance q_h, as a function of coolant mass velocity G, the primary insulation R_w, and the coolant temperatures T_f. It is evident that G has but a secondary influence on q_h, especially for $R_w > 2$ (hr-deg F-ft^2 of hot wall)/Btu, and $G > 1.5$ lb/(sec-ft^2). It is also evident that the magnitude of R_w largely controls q_h.

The energy transfer to the coolant air q_v, as given in Fig. 3, is controlled primarily by the magnitude of R_w, but here the coolant temperature T_f, has a significant influence.

The heat leak to the machinery space q_∞, as shown in Fig. 4, is greatly reduced from the q_h magnitude by the amount of energy transfer to the coolant q_v. The coolant temperature T_f has controlling influence on q_∞, and as before, for $R_w > 2$, and air flow $G > 1.5$ lb/(sec-ft^2), the influence of G is small.

Fig. 5 presents the temperature of the outer surface T_o, as a function of R_w, T_f, and G. This temperature is controlled primarily by the coolant temperature, and (for $G > 1.5$ lb/(sec-ft^2), $R_w > 2$) is approximately half way between the coolant temperature T_f, and the machinery-space temperature T_∞, with the outer blanket used.

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[4] "Heat Transmission," by W. H. McAdams, McGraw-Hill Book Company, Inc., New York, N. Y., 1942.

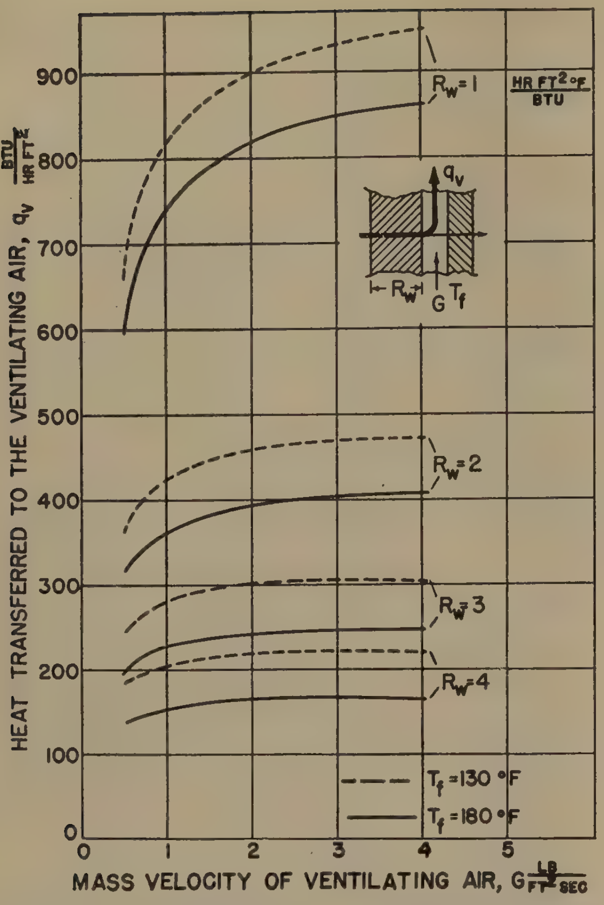

Fig. 3 Heat Leak to Ventilating Air, Variation With Mass Velocity of Ventilating Air and Primary-Insulation Resistance

(Hot-surface temperature, 1200 F; machinery-space temperature, 90 F.)

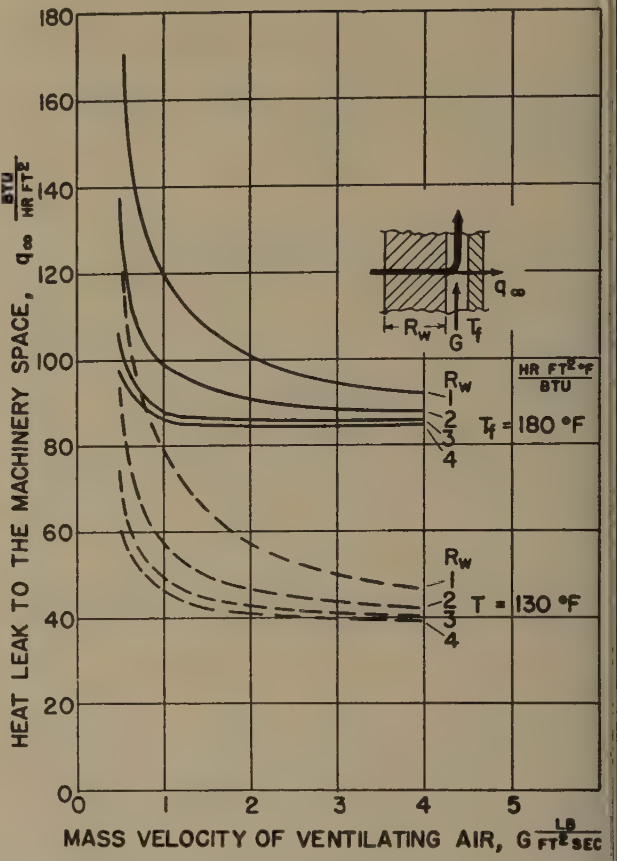

Fig. 4 Heat Leak to Machinery Space Variation With Mas Velocity of Ventilating Air and Primary-Insulation Resistance

(Hot-surface temperature, 1200 F; machinery-space temperature, 90 F.)

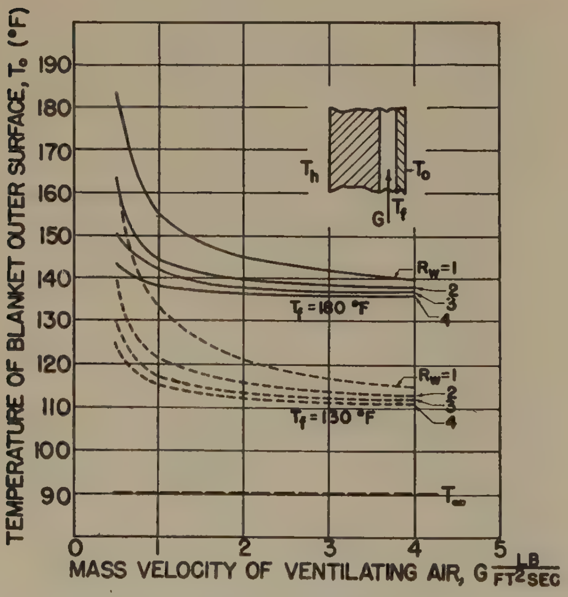

Fig. 5 Outside-Wall Surface Temperature, Variation With Mass Velocity of Cooling Air and Primary-Insulation Resistance

Hot-surface temperature, 1200 F; machinery-space temperature, 90 F.)

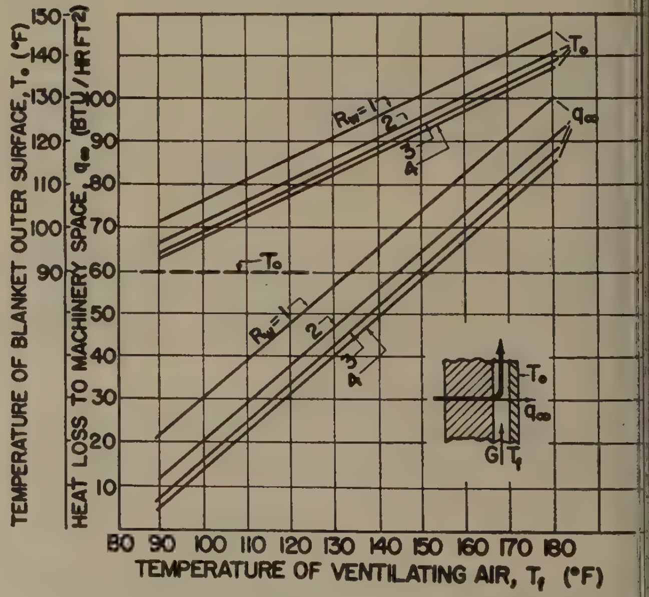

Fig. 6 Outside-Wall Surface Temperature, and Heat Lea to Machinery Space, Variation With Ventilating-Air Temperature

(Mass velocity of ventilating air = 2 lb/(sec-ft²); hot-surface temperatu 1200 F; machinery-space temperature, 90 F.)

Fig. 6 was prepared for a coolant flow $G = 2$ lb/(sec-ft²), which is considered as a typical magnitude (velocity = 30 fps for T_f = 130 F). As T_f increases approximately linearly with flow length, this graph exhibits the expected variation of T_o and q_∞ from the ventilation-air inlet to the outlet section. For instance, if the coolant flow is induced from the machinery space at 90 F and is discharged at 150 F, for $R_w = 3$, T_o varies linearly from 94 F to 124 F, and q_∞ also varies linearly from about 8 to 60 Btu/(hr-ft²). Reference to Fig. 3 will show that, for these conditions the ventilating-air energy gain is about q_v = 310 Btu/(hr-ft²). If the ventilated-space dimension δ, is specified, the foregoing data can be used to determine the flow length, l. From energy-balance considerations

$$G \times 3600 \times \frac{\delta}{12} \times 1 \times C_p\ (T_{f\text{out}} - T_{f\text{in}}) = l \times 1 \times q_v$$

Thus l = 15 ft for a ventilation space dimension $\delta = {}^1/_2$ in.

To illustrate the type of design information which can be extracted from Figs. 2 and 6, suppose it is desired to predict the performance of a ventilated thermal insulation consisting of the following:

Primary-insulation layer, with k = 0.667 Btu/(hr-ft²-deg F/in.) .... 2 in. thick (flat wall)
Ventilated space .... δ = 0.50 in.
Secondary-insulation blanket, with k = 0.55 Btu/(hr-ft²-deg F/in.) .... 0.25 in. thick
Over-all insulation structure thickness = 2.75 in.
Hot-wall temperature .... T_h = 1200 F
Room temperature .... T_∞ = 90 F
Coolant temperature .... $T_{f\text{in}}$ = 90 F
$T_{f\text{out}}$ = 150 F
Coolant mass velocity G = 2 lb/(sec-ft²)
From the foregoing information R_w = 3 (hr-ft²-deg F)/Btu, $T_{f\text{avg}}$ = 120 F

Then from the curves:

Heat leak from working substance.. q_h = 345 Btu/(hr-ft²), (Fig. 2)
Heat transfer to cooling air, average .... q_v = 310 Btu/(hr-ft²), (Fig. 3)
Heat leak to machinery space, (Fig. 6):
  At air-inlet section .... q_∞ = 8 Btu/(hr-ft²)
  At air-outlet section, ... q_∞ = 60 Btu/(hr-ft²)
  Average .... q_∞ = 35 Btu/(hr-ft²)
Calculated flow length .... 15 ft

This last result indicates a coolant-flow requirement of 5.56 lb/(sec-1000 ft² of hot-surface area) or 4500 cfm/1000 ft² of 90 F inlet air at atmospheric pressure. It is estimated roughly that the hot surface to be insulated in a high-efficiency gas-turbine plant should not exceed 300 sq ft per 1000-shp rating. Consequently, the coolant-flow requirement is seen to be only about 1300–1400 cfm per 1000 shp. Elementary calculations indicate that the pressure-drop will be on the order of 2 in. of water for the 15-ft flow length. The resultant blower-power requirement then amounts to less than 1 hp per 1000-shp rating of the gas-turbine plant. It is believed that these ventilating-air requirements are not excessive.

A conventional insulation structure may be designed for a heat eak from the working substance of the order of 150–200 Btu/(hr-ft²). Consequently, in comparison, the ventilated structure may be charged with an extra heat loss of about 200 Btu/(hr-t²). With this information, and taking the previously used 300 sq ft of hot-wall area per 1000 shp, together with a plant thermal efficiency of 25 per cent, the cost in plant specific fuel conumption may be estimated as 0.6 per cent.

It should be observed that a stressed pressure-vessel wall can be located within the insulation structure to control its temperature. Since the thermal resistance of the metal wall is small, the effect on the performance of the insulation structure is negligible. Many possible variations of wall location can be visualized for specific applications, such as combustion-chamber walls, ducting, and heat-exchanger shells. Such designs may prove useful in minimizing thermal-expansion problems.

As mentioned previously, insulation thickness on cylindrical surfaces must be greater than on plane surfaces, for the same thermal resistance expressed (hr-deg F-ft² of hot wall)/Btu. An elementary analysis demonstrates that

$$\frac{z_{\text{cyl}}}{z_{\text{slab}}} = \frac{D_o/D_i - 1}{\log_e D_o/D_i} = \frac{z_{\text{cyl}}}{\dfrac{D_i}{2}\log_e \dfrac{D_o}{D_i}} \qquad [10]$$

Fig. 7 is a plot of this relation in the form of z_{slab} versus z_{cyl} for various nominal pipe sizes from 6 to 30 in. To illustrate the significance of this effect on the dimensions of the insulation structure for a given performance, consider that a 6-in. pipe is to be insulated to give the same performance with respect to q_h, q_v, q_∞, and air-coolant temperature change ($T_{f\text{in}} - T_{f\text{out}}$) as the previously considered flat-wall system. From Fig. 7 the thickness of the primary-insulation layer must be increased from 2 in. to 3 in. The secondary-insulation layer must be increased from 0.25 in. to about 0.35 in. to give the same R_o (hr-deg F-ft² of hot wall)/Btu. The air-flow length l, however, will remain about the same, as G may be reduced to maintain the same coolant flow per unit hot-wall area. The net result is that the 6-in. pipe requires a structure of 3.6 in. in thickness as compared to 2.75 in. for the plane surface.

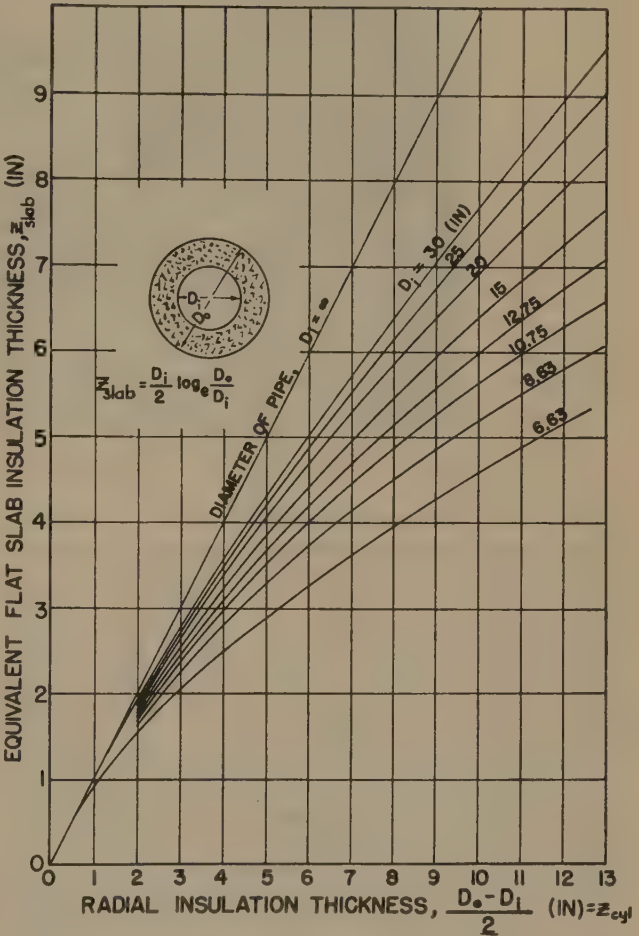

Fig. 7 Flat-Wall Equivalent of Cylindrical Insulation Thickness

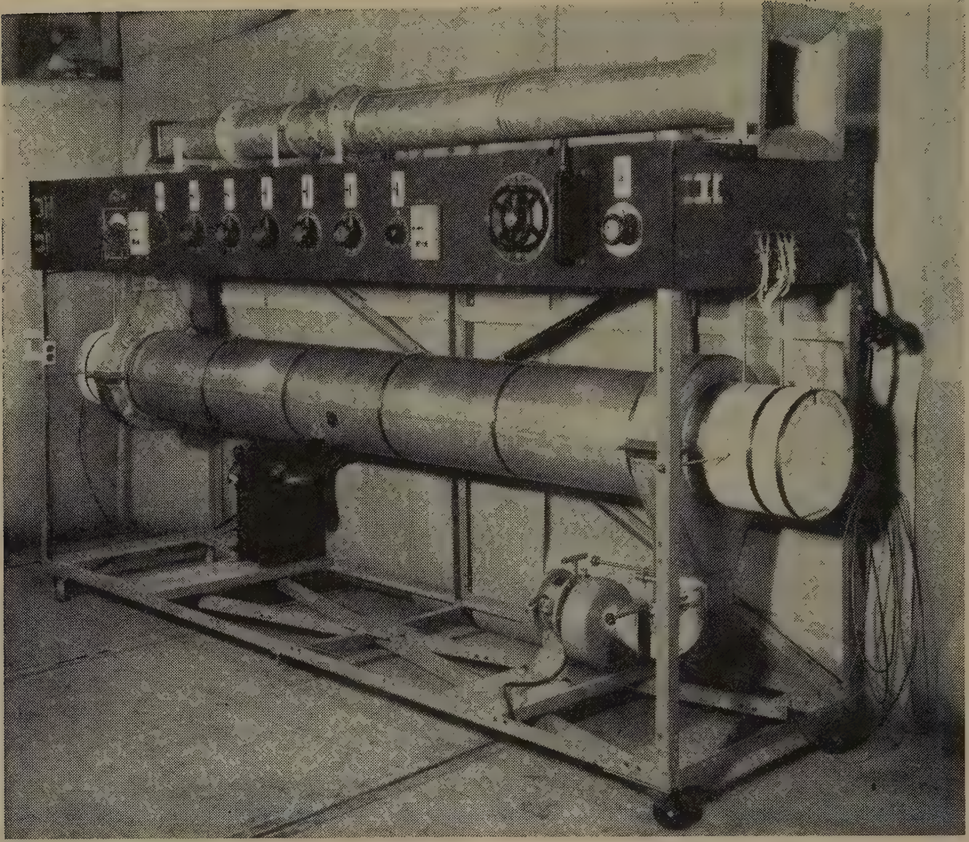

Fig. 8 Test Setup for Determination of Performance of Ventilated Insulation Structures

Experimental Results

The experimental results presented here are the first of tests currently in progress using the apparatus shown in Fig. 8. This equipment consists of an 11-ft 6-in. length of nominal 6-in-diam pipe, insulated as illustrated in Fig. 9. Note that the $^1/_2$ in. of asbestos secondary insulation has been removed from the system, as shown in Fig. 8, in order to show the heat meters used on the aluminum shell of the ventilated annulus. An electrical heating element located within the pipe provides the thermal current, q_h. The cooling-air flow is induced through an orifice meter and thence through the insulation by a small blower. The ventilating-air heat gain q_v, was established from measurements of air-flow rate, inlet-air temperature, and exit-air temperature. The three heat meters,[5] located at the mid-position of the test-section length on the outer shell of the ventilated annulus, as shown in Fig. 8, were used to measure an average q_∞ for the test section.

Compensating heaters were used to establish zero temperature gradient along the pipe at the end sections. Thermocouples located on the pipe and at the various layers of the structure provide a complete picture of temperature conditions.

Comparison of the sum $q_v + q_\infty$ with electrically measured q_h revealed energy-balance discrepancies of the order of 1 per cent for steady-state conditions, giving confidence in the accuracy of the measured test performance.

Fig. 9 shows the predicted magnitudes of the component resistances. Employing this information, performance was predicted as shown in Table 1. Test results obtained with $T_h - T_\infty$ = 1250 F, and extrapolated to $T_h - T_\infty$ = 1300 are also shown for comparison.

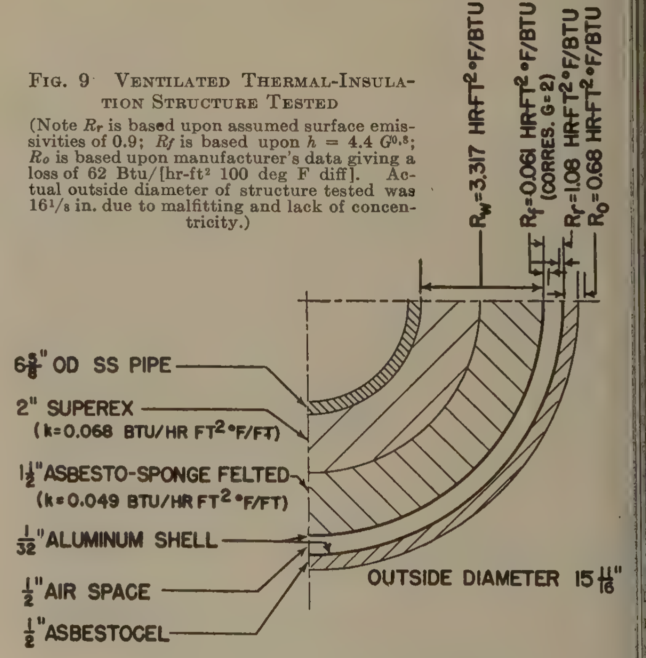

Fig. 9 Ventilated Thermal-Insulation Structure Tested

(Note R_r is based upon assumed surface emissivities of 0.9; R_f is based upon $h = 4.4\ G^{0.8}$; R_o is based upon manufacturer's data giving a loss of 62 Btu/[hr-ft² 100 deg F diff]. Actual outside diameter of structure tested was $16^1/_8$ in. due to malfitting and lack of concentricity.)

The excellent agreement is evident, with the exception of th q_∞ comparison. Here, however, it is believed that the predicte

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[5] "An Investigation of Aircraft Heaters V—Theory and Use of Heat Meters for the Measurement of Rates of Heat Transfer Which Are Independent of Time," by R. C. Martinelli, E. H. Morrin, and L. M. K. Boelter, NACA Wartime Report ARR December, 1942 (see also ARR 4H09 and ARR 4107, issued as Wartime Reports).

'ABLE 1 PREDICTED AND TEST PERFORMANCE FOR INSULATION

	Predicted	Test
Ieat leak from hot wall (average over test section) q_h Btu/(hr-ft^2 of pipe wall)	382	383
Ieat leak to ventilating air (average over test section) q_v Btu/(hr-ft^2 of pipe wall)	362	376
Ieat leak to machinery space (average over test section) q_∞ Btu/(hr-ft^2 of pipe wall)	20	11.3
)utside-surface temperature, above room temperature (average over test section) $T_O - T_\infty$, deg F	4.5	3.0
'ooling-air temperature rise (10.5-ft test section flow length) $T_{f\text{out}} - T_{f\text{in}}$, deg F	25	25.3

nagnitude is in error for the main reason that the magnitude f R_o was based upon the manufacturer's figure of a heat loss for he asbestocel of 62 Btu/(hr-ft^2-100 deg F temperature difference) vhere the temperature difference is measured from the surface eing insulated to the room condition. Undoubtedly, this figure s based on a temperature difference on the order of 100 to 200 F. n the tests, however, the corresponding temperature difference mounted to only 13 deg F (avg). As a consequence, radiation nd convection transfer coefficients within the cells of the mateial and from the outside surface were probably lower, and the ffective resistance R_o greater than used in the prediction. Anther factor contributing in minor degree to the difference between redicted and actual q_∞ is that the predicted value of R_r was ased upon a wall emissivity of 0.9. With the aluminum surfaces ctually used, a much lower emissivity and higher R_r existed. Iowever, since $R_f << R_r$ for the system and mass velocity ested, the influence of R_r was minor.

This structure has an over-all thickness of $4^3/_4$ in. However, y using a primary-insulation-layer material with $k = 0.0467$ Btu/(hr-ft^2-deg F/ft), an amosite asbestos felt of approximately 0 lb per cu ft density, the thickness of the structure can be reluced to $3^5/_8$ in., a saving of $2^1/_4$ in. on the diameter, for the same erformance.

SUMMARY AND CONCLUSIONS

1 A ventilated thermal-insulation structure can be made for nsulation of high-temperature surfaces which will have the charcteristics of small over-all thickness, low heat leak to the mahinery space, and high heat leak from the working substance, vhen compared with the conventional types of insulation.

2 For a gas-turbine plant, the high heat leak from the working substance may not be excessive. The blower-power requirenent and size are small. The number and size of air ducts are ot excessive.

3 A method for predicting the performance of the ventilated tructure has been presented, and the predictions have been onfirmed by the limited experimental data so far obtained.

ACKNOWLEDGMENTS

The work reported in this paper was made possible by the J. S. Navy Office of Naval Research and the Bureau of Ships. 'he authors are appreciative of the assistance provided by Mr. ack Reed, graduate student, and Mr. H. J. Jesperson, research ssociate, in preparation of calculations and design of the test ystem.

Discussion

E. A. RICHARDSON[6] AND G. A. RICHARDSON.[7] The theoretical nd experimental investigation of the ventilated form of insulated nclosure deals with one possible form of control of heat loss from he working substance, heat limitation to the outside of the enlosure, and protection of structural members. In the proposal, a pipe carrying gas at 1300 F is insulated on the outside by primary insulation, while cooling gas is circulated in the passage formed between the primary, and the secondary insulation. It is noted that the rise in cooling-fluid temperature is about 60 deg F in the example given and that the heat so picked up is wasted.

The concept of effective thermal isolation is not new, in fact the partners investigated means for securing equivalent results nearly 15 years ago. The insulation of boilers in steamer holds to prevent excessive heat leakage and cooling requirements was recognized as desirable. It was seen that progress of all high-temperature high-pressure processes would depend upon thermal isolation combined with recovery of heat lost from the working fluid.

One example of such an investigation was the insulation of the walls of a petroleum cracking chamber operating at 850 F and under 750 psi pressure. A slightly permeable brickwork was conceived placed inside of, and slightly spaced from the shell, while fluid was pumped into this space at such rate as to secure substantially atmospheric temperature in the shell. Using oil to be cracked, it was possible to pump 1 or 2 per cent of the process oil through the permeable layer, whereupon the "cooling" fluid would be heated to process temperature, recovering virtually all of the heat normally lost. As the steel wall no longer involved creep behavior and no longer had a temperature gradient, it became possible to use about one-half as much flanging-grade steel in the shell as of the somewhat alloyed and stainless-clad steel actually used, while securing a much safer design.

It becomes of interest to see how the partners might propose to handle the case discussed by the authors of this paper. But first, it must be realized that the partners had removed the temperature limit to cracking processes in so far as structural materials might be concerned and imposed temperature limits, if any, on the basis of refractory behavior.

Retaining the "Superex" and asbestos-sponge felt shown in Fig. 9 of the paper, the partners might substitute $^1/_2$ in. of glass wool for the asbestocel layer and remove the aluminum shell immediately within this layer. They would then cause air to be passed radially inward through the glass-wool layer into the ventilating space, the air to be exhausted therefrom as at present. Suppose the entering-air temperature is 80 F while the outer surface of the glass wool is at 90 F. The heat loss from the outer surface cannot exceed 5 Btu per sq ft per hr for this 10 deg F drop, for, obviously, convection cannot occur.

Having set the outer-surface temperature 10 deg F above the entering-air temperature, we can assume a set of values of hot-surface temperature of glass wool, and determine therefrom the value of nC_pL/k_s, where n is the rate of fluid permeation in pound-mols per square foot per hour, C_p is the molal specific heat in Btu per 1 deg F per lb-mol, L is the insulation thickness (normally in feet, but for this case in inches), and k_s is the effective conductivity of the glass-wool and air combination. It is then possible to tabulate the value of hot-surface temperature as T_h, the other quantities given, and add data on the heat to the air and the required temperature of the outer surface of the primary insulation to supply this heat by radiation to the glass wool, convection being neglected.

TABULATION FOR FIRST SUBSTITUTION

T_h	nC_pL/k_s	C_p	k_s	n	q	T_{pi}
500	3.76	6.91	0.480	0.523	1528	738
400	3.47	6.88	0.444	0.447	985	614
300	3.10	6.85	0.413	0.373	563	479
200	2.49	6.83	0.386	0.281	230	320

NOTE: Term k_s is in Btu per hr per 1 deg F per in.; q is in Btu per sq ft per hr; T_{pi} is temperature, deg F, outer surface of primary insulation.

If the heat leak from the hot wall is of the order of $382 = q_h$, then T_h might be 245 F by interpolation, $T_{pi} = 390$ F, and 0.32

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[6] Partner, Edward, (Nellie), and George Richardson, Bethlehem, 'a. Mem. ASME.

[7] Partner, Edward, (Nellie), and George Richardson, Bethlehem, 'a.

lb-mol of air per sq ft per hr are required. This is 9.25 lb per sq ft per hr. For the 15-ft length of pipe, and the annular space as given, this corresponds to G = 1.08 lb per sq ft per sec, or about one half of that required by the authors. However, the air is now heated from 80, to 245 F, or 165 F, a value much larger than the 60 F in one example, or the 25 F in the test case. A decided reduction in cooling air has occurred. Furthermore, the heat loss from the outer surface is approximately constant and negligible, and not rather widely varying lengthwise as in the authors' case. Our method does not permit convection from the outer surface.

Actually, we should feel that such a method of treatment is makeshift in character. We should prefer to place a primary layer of permeable insulation inside of the pipe, as well as the annular space and the secondary insulation. In such case, the cooling air would be bled from the compressor, dividing it into two parts. One part would be caused to enter the annular passage, after a slight pressure boost, at a temperature of the order of 400 F, while another part would be cooled to room temperature for passage through the secondary insulation. A slight pressure boost of the bled air is required to insure flow at a proper rate through the permeable layers. The whole of the bled air, after it has served its purpose as a coolant, mixes with the hot gases passing to the turbines. The effect is that radiation has been almost entirely eliminated at the cost of a slight boost in pressure of the bled air, while the temperature limit of the cycle, in so far as piping is concerned, has been eliminated. It would be just as practicable to use gases at 3000 F as at 1300 F. The gases not bled are at a slightly higher temperature, leaving the combustor, than would be normal. The pipe is at atmospheric temperature. Although larger in diameter, it can be lighter in weight and safer than the stainless-steel piping shown by the authors, as well as much less costly.

Assume the value of k_s (per ft, not per in.) as 1.0, the mean specific heat of air between 425 and 1300 as 6.70, and an outer temperature of primary layer 3 per cent of the difference of 875 F (26 F) above the entering air temperature, the value of nC_pL/k_s becomes 3.50, so nL = 0.52 (L now in ft). Using L = 0.25 (3 in.), then n = 2.08 lb-mol per hr per sq ft of 400 F bled air. Since the $^1/_2$-in. glass-wool layer requires (see table in foregoing discussion) about n = 0.45 lb-mol per hr per sq ft, this is the amount of bled air which must be cooled from 400, to 80 F. Thus 0.45 mol cool air, 1.63 mol hot air are bled per hr per sq ft. The heat removed from the bled air (0.45 mol) is 1100 Btu per hr per sq ft. The total heat removed from the hot gases is 18,300 Btu per hr per sq ft, fully returned in the "cooling" air. However, these results are excessive, in that it is assumed the hot gases can deliver 18,300 Btu per hr per sq ft. Actually, the inside of the primary insulation will be cooler, hence correspondingly less air will be required.

Figures used herein were derived by the partners, partly theoretically, partly by test. The rapid variation of k_s for glass wool tends to confirm the difference in heat leak to the machinery space (20 predicted, 11.3 on test) found by the authors. However, knowing the rapidity with which radiation effects increase with temperature for wool and felted insulations, we would question the statement regarding the saving in primary-insulation thickness through the use of amosite asbestos felt of 10 lb per cu ft density. Actual measurements of conductivity are required in the region above 1000 F, since extrapolation can be hazardous.

We think that it will be obvious that more promising methods of thermal isolation are available than the ventilated type of insulation investigated by the authors. The partners have done much work investigating the use of "permeable plates with fluid permeation between faces" for insulation, cooling, heat transfer, and the like, for a wide range of applications.

AUTHORS' CLOSURE

In considering means of controlling the heat loss from high-temperature surfaces the possibilities of using the thermal-energy storage capacity associated with a mass transfer through a porous material, as proposed by the discussers, should certainly be studied. The idea has been proved in many practical instances. The ordinary kitchen watercooler jug allows water to seep through the porous surface to the outside, where the evaporation provides a substantial cooling in dry climates. The combustion-chamber liner of the aircraft turbojet engine, such as the German Jumo 003, is provided with a large number of small vents through which cooling air is introduced, sweeping the inside of the liner with a blanket of cool air, and effectively isolating the low-carbon-steel outer casing (of sheet metal) from the high-temperature zone. It has been demonstrated that cooling of a rocket nozzle by the passage of a gas or a liquid through a porous material into the nozzle stream has attractive possiblities for maintaining safe temperatures in the nozzle wall.

The possibility of placing stressed parts of the machinery, such as the hot gas-duct wall, on the cold side of the insulating material is recommended by Messrs. Richardson. The possibility of doing this was mentioned in the paper, and should certainly be studied for any specific insulation application.

Certain practical difficulties are apparent in attempting to apply the specific recommendations of the discussers. In placing the primary insulation layer on the hot side of the duct wall, it still will be necessary to use a stainless liner material to hold the insulation material, which is usually of very fragile material, in position; otherwise damage might occur to any turbine machinery which might follow (especially if a ceramic material were used for the insulation). Also, mechanical protection of the cold side of the insulation, adjacent to the cooling-air annulus, would be necessary. The mechanical structure would become somewhat complicated.

It is to be noted that the stainless-steel pipe shown in Fig. 9 of the paper was not intended to represent a hot gas duct; it represented, rather, a means of securing a hot surface for the test setup which would maintain its shape and not scale excessively at the high temperatures tested.

The suggestion for using a $^1/_2$-in. layer of glass wool for the outside insulation blanket (in the machinery space) also meets with the objection of mechanical vulnerability. It would have to be supported with perforated metal both inside and outside. A dirty atmosphere also might seriously impair the porosity; this may be a valid objection on shipboard, even though the atmosphere is relatively quite clean. The thermal isolation possible with the discussers' proposal is obviously greater than that possible with the ventilated arrangement discussed in the paper.

It is not clear from the discussers' comments what the significance of the term nC_pL/k may be, nor how it is to be determined or utilized. By considering an energy balance at a point on the duct surface at thermal equilibrium

$$n \cdot C_p \cdot \Delta t \cdot \delta A \quad = \quad U \cdot \delta A \cdot \Delta t$$

associated with cooling air mass transfer	associated with conduction through insulation material

If the convection resistances at the two surfaces of the insulation are neglected, this expression reduces to

$$nC_p = \frac{k}{L}$$

and it would appear that the value of nC_pL/k should be about 1.0, rather than the value of 3.50 given. Of course much depends upon the significance of k when there is a flow of air through the insulating material.

The Mechanical Cleaning of Fouled Heat-Exchanger Tubes

By A. JOHN,[1] LONG ISLAND CITY, N. Y.

For many years in the past, mechanical cleaning of badly fouled heat-exchanger tubes was more or less done on an improvised basis. As heat exchangers began to be extensively used, fouling became a problem of major and varied proportions. This created an impetus for the tube-cleaning industry to concentrate on the development of a special tool for this purpose. Today, there is available to the trade a single-purpose mechanical cleaner suitable for almost every possible tube-cleaning requirement, as the author explains in this paper.

FOR many years in the past, mechanical cleaning of badly fouled heat-exchanger tubes was done on an improvised basis. The methods were limited to the handiest tools readily available which usually consisted of drills of various sorts or standard boiler-tube cleaning equipment. The drills approached the power problem from an external application while the boiler-tube cleaners approached the power problem internally. Boiler-tube cleaners, the motors of which enter the tube, were found rather lacking in power when heavy deposits and plugged conditions prevailed. The improvised drills, while doing a job more rapidly, generally produced mediocre results, although some methods do deserve honorable mention. In spite of the various methods and the results, information was not generally made available to others with similar problems. Whenever a problem existed and was more or less solved, it was generally confined within the plant or at best, the information was passed along on a word-of-mouth basis. Undoubtedly, many maintenance men used similar methods but unknown to each other, the knowledge of which if pooled together would have been of inestimable value.

When tubes are not badly fouled or not plugged other methods than strictly mechanical ones have been used. Although those in the trade do not necessarily agree that sandblasting, shooting of rubber plugs, pickling and steam, hydraulic or air lancing comes within the scope of mechanical cleaning, a word in this respect will, nevertheless, be of general interest. These methods have been tried at various times and in many cases are superior to mechanical methods. For instance, in a bank of condenser tubes, where deposits are negligible or of the muddy or river-growth type, the practice of shooting rubber plugs is superior both in speed and performance. This method is only for the removal of soft deposits and obviously will not remove hard scale. The other methods mentioned have been used successfully at various times, and some of them are the only ones available for the cleaning of the shell side of heat exchangers.

Obviously, prior to the consideration of any mechanical cleaning method, hand methods with brushes and scrapers were also involved. The hand method may still have its occasional place but it can be considered as outmoded in heavy industry.

During the past 10 years, many additional plant processes have been developed utilizing heat exchangers more and more extensively. Thus the incidence of fouling troubles greatly increased and, since production was a byword, its incentive applied to tube cleaning as well. This created the impetus to concentrate on the development of a special mechanical tube cleaner, particularly one to remove very heavy scale deposits and also to open up plugged tubes. Many improvements of mechanical cleaning methods manifested themselves during this period and, ultimately, a high-speed direct-drive scavenging drill-type cleaner was successfully developed and marketed.

Fouling and Characteristics of Scale

The heat exchangers to be cleaned are generally located in process industries such as oil refineries, fruit and vegetable juice, synthetic rubber, plastic stock, paper pulp, dairy, and miscellaneous chemical plants. The tubes of these heat exchangers are predominantly straight and without return bends, although some tubular-return types exist. The majority are mounted in a horizontal manner providing a convenient approach from the cleaning viewpoint. Commercial tube sizes range from $^1/_2$ in. OD to 2 in. OD with the $^5/_8$ to 1-in-OD range predominating. Lengths vary from 18 in. up to 30 ft and occasionally longer.

Vertical installations also exist such as the open-head shell-and-tube condensers found in refrigeration plants and sulphite evaporators in pulpmills. Although not heat exchangers of the liquid type, air preheaters, as installed in boilers, can be considered as coming within the scope of this paper. The tube lengths of vertical installations generally run up to approximately 50 ft. Where such air heaters are installed horizontally, the tube lengths average approximately 8 ft.

The fouling of heat-exchanger tubes is generally caused by localized high temperatures, precipitation of solids at high or low temperatures, low velocity of flow through tubes, cutting units off stream with the normal settlement of solids, and the congealing reaction of certain liquids. In respect to boiler air preheaters, the most common cause of serious fouling is a change of fuel firing, such as from coal to oil or vice versa without time taken out for cleaning. The small amount of gummy residue, deposited when burning oil, combining with fly ash ultimately tends to form into a solid mass. However, it is believed that the lack of periodic preventive maintenance can be considered as the greatest cause of the complete fouling of the majority of heat exchangers.

The type of scale deposited covers a wide range from flint hardness to soft and gummy. Interspersed are conditions such as (in all ranges of hardness), porous or cellular, compacted but elastic, adhesive or gummy, blistery, etc. Some scales adhere to the metal walls with such tenacity as to be difficult to remove cleanly, while some are readily dislodged once the internal pressure of being compacted is removed. A myriad of chemical compositions exists, such as, calcium compounds, silicates, carbonates, sulphates, polymers, coal tar, solidified fatty acids, miscellaneous chlorides, algae and other organic matters, as well as common iron oxide.

Each type of scale presents a different problem of removal, especially as to the proper combination of tools and methods. Past experience is also a determining factor and must not be overlooked.

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[1] Assistant General Manager, Thomas C. Wilson, Inc.

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society. Paper No. 48—A-114.

The Direct-Drive Air-Operated Mechanical Tube Cleaner

One tube cleaner, which has been developed and has proved very popular, is known as the outside-suspension type and in general appearance resembles an electric drill. It consists of a high-powered air motor operating outside of the tube bundle, transferring power through a hollow rotary shaft to which is attached a special drill bit, brush, or other suitable accessory. The hollow rotary shaft provides a means of introducing a scavenger such as air, water, or steam under pressure direct to the point of cutting and serves to soften the fouled matter, wash away the debris, or to keep the cutter bit cool. Accessories are devised so as to allow passage and exhaust of the scavenger medium as close to the point of cutting as possible. This cleaner has proved to be a boon to many harassed plant engineers and, due to lightweight and portability, it provides an "on the spot" method of removing scale and other fouling matters rapidly. This type of cleaner also provides the only effective method that can be used to clean tubes which are plugged solidly.

Fingertip control is provided for a variety of speeds, and at present two types are manufactured. A light-duty unit suitable for one-hand operation is available weighing 3 lb, and a heavy-duty unit is also manufactured which weighs 17 lb. The same accessories can be used with either cleaner within the similar size ranges. The light-duty cleaner is generally used on short-tube heat exchangers from $^{1}/_{4}$ in. ID to $^{7}/_{8}$ in. ID. The heavy-duty cleaner is suitable for tubes as small as $^{3}/_{8}$ in. ID, and up to 2 in. ID. Under ideal light-scale conditions, and where production warrants the application, or the equipment happens to be available, these ranges can be somewhat extended. For instance, it is recalled that the heavy-duty cleaner was used to brush out a quantity of 3-in-OD straight tubes which had become rusted or otherwise fouled to a moderate extent.

The hollow rotary shaft of the cleaner is made in lengths to suit the heat exchanger being cleaned. Where heat exchangers are located close to walls or otherwise are obstructed and sufficient headroom is not available, shafting can be supplied in short lengths and coupled together as progress is made. The normal flexibility of the shafting, in any length, is sufficient to follow the contour of sagged tubes. The cutter bit is usually a rear-piloted type and, coupled with the limberness of the shafting, normally stays within the confines of the inside diameter of the tube. A slight whipping action is perceptible at the highest rotary speeds when using long shafting. This action has no bearing on the results or the normal operation of the cleaner, but there is evidence that the possible contact of the shaft with the tube wall has a tendency to remove residual scale deposits which may have been left after passage of the cutter bit.

Normally, constant forward pressure on the cleaner is satisfactory during cleaning. However, occasionally when flintlike deposits prevail, a vibratory attachment can be used which greatly facilitates progress. This attachment, used with the heavy-duty cleaner only, creates a staccato of sharp blows, insufficient in intensity to cause buckling of the shafting but intense enough to dislodge scale in sizable chip form. Blows average approximately 1900 per min and occur simultaneously with rotation. An analogy can be made to the well-known air-operated rock drill or paving hammer. The vibrating attachment also reduces the laborious aspect of a difficult problem by approximately 60 per cent. Records indicate that approximately 15 per cent of the cleaners sold are provided with this attachment.

Drill bits are of a special-compounded alloy steel capable of sustained operations even under temperatures approaching 600 to 700 F. They are generally three- or four-fluted in design and ground at a cutting angle of approximately 120 deg. The cutting edges are not keen but rather stubby and are ground with reverse rake angles. Although this cutting edge is standard, it has been found that many operators grind the bits to suit thei personal whims or what is compatible with the cleaning operatio being performed. Occasionally on some plugged tubes auge type bits are used when the deposit is spongy and of a rubbe like consistency. Records indicate that very seldom are cond tions alike plant to plant, and some allowances must be mad and time allotted for proper analysis. However, on hard deposits, the fluted style of bit has proved the most popular. Whe tubes are not plugged and deposits are soft or powdery, expanding brushes or expanding cleaners of a scraper type can be substituted satisfactorily.

The selection of a suitable scavenging medium is also importan but, as is quite often the case, it must be what is available at th plant. Mainly due to their general availability, air or wate seem to be the favorite scavenging mediums. Occasionally, ho water and steam are used, particularly on deposits which ma soften under high temperatures. However, steam has the disadvantage of condensing rapidly, thus reverting back to a ho water medium. Condensation is particularly rapid when lon tubes are encountered and the steam source is a considerable distance away.

It is recommended that scavenger pressure be high enough t overcome forward thrust or feed pressure of the cleaner. If th scavenger pressure is not higher than the feed pressure the exhaust holes in the drill bits may become clogged, particularly the tubes are plugged. It has also been ascertained that whe tubes are plugged, the smaller the tube being cleaned the greate the scavenger pressure required. This is necessary in order t eliminate cutter-bit plugging and also because of the physica limits of the rotating-shaft diameters. In order to allow th debris to be forced back toward the tube mouth the outsid diameter of the rotating shaft must of necessity be smaller tha the inside diameter of the tube being cleaned. This restrictio limits the volume of scavenger under moderate pressures an consequently, to provide for a rapid exit of debris, pressures mus be increased. When tubes are not plugged pressure may be lowe and in many cases a trickle of water at city pressure will was out the debris effectively. Records indicate that water pressure as low as 30 psi and as high as 225 psi have been used effectivel Air pressure is usually limited to the available source which i most cases is 125 psi. It should be noted that no definite ru can be set, and recommendations are generally made from experiences of the past.

Once the proper method is established, it should be recorded fo future use, thus eliminating further experimentation by th cleaning group. Also, periodic cleaning schedules may be established in order to eliminate sudden process stoppages and i many cases eliminate retubing.

Various Cleanouts and Results

The actual operation is rather a simple and convenient one, a can be noted from the illustration of the heavy-duty unit in Fig. This view shows a heat exchanger approximately 15 ft in lengt with 1-in-OD solidly plugged tubes. The deposit was fair hard but the tubes were cleaned at the rate of 3 min each. A though Fig. 1 shows this heat exchanger removed from the lin this practice is seldom carried out. Seldom do reasons exist fo not cleaning the heat exchanger in the installed position. O of the fastest recorded times, with a similar heat exchanger, o curred in a Michigan chemical plant. The tube bundle was sim lar to that in Fig. 1, but a slightly different scale deposit wa found. The deposit was hard but not compacted and althoug the cutter bit progressed rapidly, the debris was not exhauste satisfactorily with air scavenging. Cold water at a pressure 140 psi was tried, with the result that each tube was cleaned $1^{1}/_{4}$ min.

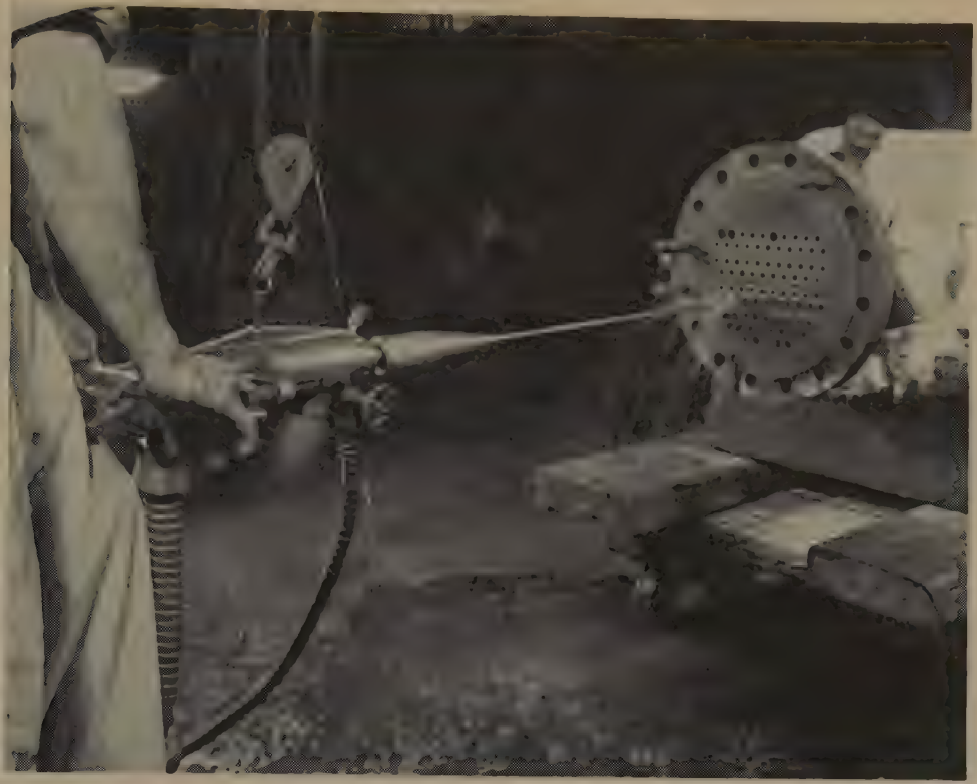

Fig. 1 Heavy-Duty Tube Cleaner

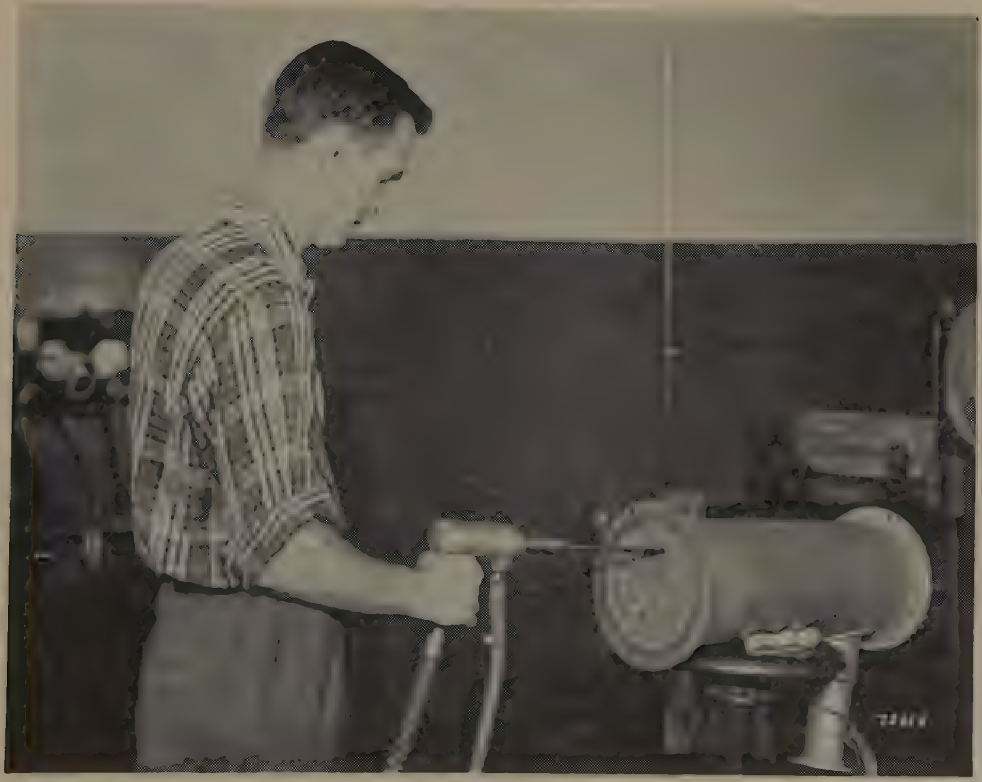

Fig. 2 Light-Duty Tube Cleaner

Recently a battery of approximately eight heat exchangers at a Pennsylvania refinery became solidly plugged with a flint-like deposit. The cleaning crew was doing the best that could be accomplished with the existing tools. Owing to unusual conditions the best was not satisfactory, and a faster method was sought. In a general discussion, retubing was even considered as being faster than existing methods of cleaning. A most unusual condition prevailed in this battery of heat exchangers as the majority of the tubes were plugged for a distance of approximately 3 ft from either end. This was due probably to localized heat at the terminal ends, but since this condition was effective in stopping circulation, the units had to be taken off stream for cleaning.

Two heavy-duty cleaners were dispatched for this plant on a rush order. The advantages of these units were demonstrated on the spot in comparison to the cumbersome previous efforts. Mechanical cleaning in similar instances has since become a standard procedure.

The results mentioned previously are only a few accomplished with the heavy-duty cleaners. The light-duty cleaners, Fig. 2, have a similar record of performance. In a New Jersey soap plant a short heat exchanger was slowing down in performance. Records indicated that this heat exchanger had not been cleaned in approximately 8 years and that the hard water used was the cause of the fouling. The heat exchanger was mounted in a very awkward position in respect to retubing or replacement, an operation which would take approximately 1 week. The light-duty cleaner performed the cleaning operation, at an invested cost in tools of less than $125, in approximately 16 hr without seriously hampering process operations. Normally, a heavy-duty cleaner would have been recommended, and the job could have been halved in time. However, owing to the general inaccessibility of the heat exchanger, the lighter cleaner seemed more practical. It was also planned that thereafter the equipment would be cleaned periodically, a job most suitable for a small cleaner.

In a nationally known electrochemicals plant at Niagara Falls, the same type of light-duty cleaner provided a cleanout in $^1/_2$ hr, an operation which formerly took 5 to 8 hr by previous manual methods. Since this cleaning was necessary every week, the savings are obvious and, in fact, with a few cleanouts, the savings amortized the basic cleaner cost.

ELECTRICALLY DRIVEN FLEXIBLE-SHAFT TUBE CLEANERS

Electrically driven flexible-shaft cleaners have been found satisfactory for use in confined quarters or where compressed air is not readily available. This type of cleaner is also popular for cleaning certain process calandria units, and the vertical open-head shell-and-tube condensers in refrigeration plants. It may be well to point out that in an ammonia-compressor plant, enormous savings in electric power used for compression not only are possible but an actual fact. It has been ascertained that even after hand-cleaning, a substantial amount of the deposit remains, and that a follow-up with the mechanical cleaner removes enough additional deposit to reduce the ammonia head pressures between 10 and 25 psi. Taking a pressure reduction of 25 psi into consideration, a 300-ton per day ice plant, operating 300 days a year, could save approximately $7500 yearly in electrical power alone, a saving which more than pays for cleaning equipment and the labor involved during the course of a year.

When cleaning the open-head shell-and-tube condensers in a refrigeration plant with electrical equipment, no scavenging operation is necessary. The condenser can be cleaned without process stoppage and while the cooling water is in circulation. Since the circulation is downward, the debris is carried to the bottom sump and generally can be removed readily.

The case history concerning a cereal plant in a New York are is on file indicating the fact that three vertical evaporators, eac consisting of 680 2-in-OD copper tubes, 3 ft long, were cleane successfully with an electrically driven cleaner in 2 hr by tw men. This compares favorably with former cleaning of only tw evaporators in 1 day, involving the labor of four men using manua methods. Brushing the light soft malt deposit with a nylon typ of brush also produced a highly polished surface, a desirable cor dition which never existed before. Cleaning in this case is don every 2 weeks with great savings in cost over previous practices.

The electrically driven cleaners range from $^1/_4$ hp to $1^1/_2$ h and can be furnished with prevailing electrical characteristic Tubes of $^1/_2$ in. to 4 in. ID can be cleaned. Quite often remot push-button control devices are featured. Flexible shafting manufactured in standard lengths from 15 ft to 35 ft in 5-ft add tions, the most convenient length being roughly 5 ft in excess c the tube lengths being cleaned.

CLEANING RETURN-BEND-TYPE HEAT EXCHANGERS

Previous discussion on cleanouts applies particularly to nor return-bend-type heat exchangers. The necessity for cleanin the return-bend-type heat exchanger does not often exist as th majority of such installations are located in nonfouling processe If serious fouling conditions are anticipated, designers invariabl will specify that the straight-tube nonreturn-bend type of hea exchanger be installed. The U-bends of the return types are th restricting factors. They are difficult to clean, and in mos cases it is impossible to do so mechanically. However, if retur bend types do need cleaning and providing the radius is generou it is occasionally possible to do so by means of flexible attach ments. These attachments are inserted, on both the air or ele tric cleaners, between the shafting and the cutting tools an again, depending upon the radius, will completely negotiate th U-bend from one side or will clean halfway. The other half c the U-bend can then be cleaned through its corresponding se tion of straight tube. However, this is practical only if the de posit is fairly soft, the tubes are not completely plugged, or at leas not plugged with a very hard deposit. Cleaning can be by eithe the air or electrically operated units.

STANDARD AIR, WATER, OR STEAM-DRIVEN BOILER-TUBE CLEANERS

These cleaners are of the so-called "turbine" type, the moto of which are either true impulse turbines or expansion-reactio type in design. These motors are usually adjacent to the cuttin accessory, hence the cleaner must enter the tube. Feeding done by means of an appropriate hose which is resilient and plia ble, thus being able to negotiate the bends which predominat in the majority of boiler tubes. History would indicate that th water-driven turbine type was first developed approximately 4 years ago. At that time the use of compressed air was not popular reality. It must be assumed that since water pump were always available around the boiler room, water as motiv power was natural. Today the air-driven cleaner is the mo popular, and this cleaner, as well as those driven by steam c water, can be used for the cleaning of heat exchangers. Approx mately 30 per cent of the turbine type of cleaners manufa tured are for use in heat-exchanger cleaning. However, its us is limited to tubes which are not plugged, or where deposits a not very thick, and particularly where a substantial amount c time is available for tube cleaning. Quite often this time facto coupled with the lower investment of the standard boiler-tub cleaner, may be the deciding factor relative to the selection of an of the cleaners described in this paper. To some extent the cu ting accessories are similar, and there are also many cases c record where drill bits have been directly attached with successf esults. When utilizing these cleaners, the general practice is o eliminate auxiliary scavenging mediums as in most cases the xhausting water, air, or steam is sufficient to carry away the debris. These cleaners are made for the cleaning of tubes as mall as 1/2 in. ID. For all practical purposes there are no retrictions to the upper limits, and tubes up to 6 in. ID are commonly cleaned, although not necessarily heat-exchanger tubes.

Conclusions

Experience with mechanical cleaners has indicated that large avings in capital investment, as well as increased process efficiency are attainable. The usual maintenance crew is sufficienty xperienced to handle mechanical cleaners which can be procured t a very moderate cost. Consequently many plants have established this method as the preferred one, particularly for immediate "on the spot" service, and also as spare emergency cleaning equipment. Experience also indicates that these cleaners usually perform their task satisfactorily long before other methods are started. For the opening up of completely plugged heat xchangers, mechanical cleaning is the only method available barring retubing. At present there is no known method of pickling, lancing, or otherwise clearing out plugged heat-exchanger tubes.

Since the advent of the specially designed cleaner, manufacturers have noticed consistently an increased interest and demand for its use. It must be presumed that this interest has been aroused because of the feasibility of such a product and method. Unquestionably, as time progresses, the improvement of mechanical cleaners will keep pace with engineering concepts of maximum work output at a minimum invested cost, and in the shortest length of time.

Discussion

F. E. Johnston.[2] Prior to the introduction of the pistol-type hollow-shaft and cutter-bit tube cleaner, we had tried various methods of cleaning badly fouled condenser tubes using varied types and combinations of equipment ranging from twisted square cold-rolled stock hammered out to an arrowhead which was twist-driven into the tube with a hammer and then turned by a pipe wrench to loosen it.

This method was crude and time-consuming, as well as inefficient. The same type rod was next used but propelled by electric drilling machines. This procedure resulted in burned-out equipment, and very tired and disgusted operators, as well as a costly and unsatisfactory job.

Both of the foregoing operations were carried out on a dry basis at long and unscheduled intervals.

Then we procured a pistol-type tube cleaner, using air at 100 psi for driving the drill head. With water as the flushing agent at 70 psi, we were able to speed up operations and obtained a more effective and therefore a more efficient cleaning job.

As pointed out in this paper, no matter how good the available equipment, unless a definite schedule of periodic cleaning is set up and adhered to, tube cleaning will be slow, laborious, and costly.

The technique of the men performing this work is also important. They must be trained thoroughly in the use and care of the equipment and the most effective methods must be employed. An operation of this nature requires patience, skill, and conscientious application to the job at hand.

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[2] Bergenfield, N.J.

Chemical Cleaning of Heat-Exchange Equipment

By C. M. LOUCKS[1] and C. H. GROOM[2]

The use of chemical solvents in cleaning heat-exchange equipment is described on the basis of the types of deposits, the solvents available, and the metals encountered. The application of the solvents is discussed, and a number of case histories are described, illustrating chemical cleaning in various industries.

IT is logical to expect heat-exchange surfaces to experience some degree of fouling. When the fouling becomes serious enough to cause appreciable loss in efficiency of heat transfer, some measure of corrective maintenance is necessary. This paper concerns the use of chemical solvents for removing the offending deposits and restoring the equipment to satisfactory performance.

The successful and economic application of solvent cleaning depends upon a number of factors, namely, the deposits, the metals in the unit to be cleaned, the solvents available, and safe operating practices. It is proposed to discuss these factors as they influence solvent cleaning and to describe the methods of application by citing experiences with solvent cleaning in various industries.

Deposits

The successful removal of any type of deposit with solvents is naturally dependent upon the nature of that deposit. In solvent cleaning of heat-exchange surfaces, the deposits, for purposes of identification, are divided into one of three broad classes, namely, inorganic deposits, organic deposits, or mixtures of inorganic and organic deposits.

Inorganic Types. The inorganic deposits consist of a comparatively small number of specific chemical compounds, which may be designated by chemical or mineral name, and by chemical formulas (1).[3] The possible constituent metals must be limited to those cations present in the water or fluid medium or formed from the metals with which the fluid medium comes in contact. The negative ions or anions will be those present in or derived from the fluid medium.

For example, if a well water is used as a cooling medium in a system consisting of steel pipe and a heat exchanger constructed with cast-iron channel sections and copper-alloy tubes and tube sheets, one may list the possible positive and negative ions to be expected as follows:

	Metal ions	Negative radical
From water	Ca	$CO_3^{=}$ HCO_3^{-}
	Mg	$SO_4^{=}$
	Na	Cl^{-}
	K	$S^{=}$
	Fe	Oxygen, $OH^{=}$
	Mn	Silica
By corrosion attack on metal surfaces	Fe	
	Cu	
	Zn	

It would be possible to discover in the deposits such compounds as are insoluble in water under the conditions existing in the system:

Carbonate or basic carbonate of	Ca, Mg, Fe, Mn, Cu, Zn
Sulphate of	Ca
Oxide or hydroxide of	Fe, Mn, Cu, Zn
Silicate of	Ca, Mg, Fe, Mn, Cu, Zn
Basic chloride of	Cu
Sulphide of	Fe, Cu, Zn

Further, if chemicals are being added to the system for corrosion control (phosphates, chromates, silicates, etc.), or for any purpose whatever, compounds containing these anions may be found in the deposits.

Organic Types of Deposits. Organic materials may be found on water sides of tube and shell units under operating conditions where:

1 The water or steam is contaminated with oil or other organic matter.

2 Conditions are favorable for the growth of living organisms, such as slime or algae.

However, typical organic-type deposits are generally associated with the product sides of exchanger units in oil refineries (2, 3, 4), coke plants, chemical industries, etc.

Organic deposits generally consist of not one or two but hundreds and probably thousands of organic compounds. The effect of heat, pressure, moisture, and other physical chemical factors can produce a great variety of organic deposits. The organic deposit may be anything from a light volatile fluid, to a highly polymerized material or decomposed cokelike deposits which appear to be untouched by the solvents usually employed for solvent cleaning. Petroleum refineries, by-product coke plants, synthetic-rubber plants, the fermentation industry, food-processing industry, the chemical industry, and others have problems with organic deposits.

Because of the almost unlimited variety of organic deposits, identification for purposes of chemical cleaning may be generalized as follows:

1 Of vegetable origin; saponifiable, or nonsaponifiable.

2 Of petroleum origin; soluble in carbon tetrachloride, or insoluble in carbon tetrachloride.

3 Of coal origin; soluble in benzene or pyridine, or insoluble in benzene or pyridine.

4 Miscellaneous

The foregoing classifications for organic deposits are not entirely satisfactory. However, since no attempt can be made to classify these materials by their chemical identities, it becomes desirable to characterize them on the basis of some simple descriptive laboratory tests. The fundamental problem is to determine whether the deposit can be dissolved in a suitable organic solvent

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[3] Numbers in parentheses refer to the Bibliography at the end of he paper.

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of he Society. Paper No. 48—A-113.

or dislodged and dispersed in an aqueous solvent sufficiently to effect its removal.

Mixtures of Organic and Inorganic Materials: It is not uncommon to find typical inorganic deposits in water and steam systems mixed with oil or grease, with living organisms such as slime, or various other materials which entered the system accidently.

On the other hand, most organic deposits contain some inorganic material resulting from corrosion of the metals in the system. For example, in a petroleum refinery, most organic deposits are mixed with the sulphides of iron or copper.

Special chemical-cleaning procedures may be necessary for dealing with these mixed deposits satisfactorily. This will be illustrated by a specific case history.

Types of Solvents

In solvent cleaning there are many solvents available. The number of practical solvents is limited, however, by cost, by fire and health hazards, and by the corrosive nature of the solvent. The type of solvent used in chemical cleaning is greatly influenced by the metals in the unit to be cleaned. The relation of the metals to solvents is illustrated in Table 1.

TABLE 1 PROPERTIES OF METALS ENCOUNTERED IN CHEMICAL CLEANING

Ferrous Metals

Cast irons can be cleaned with any of the common solvents used for that purpose. When using inhibited acid, however, it is advisable to use lower temperature limits than for low-carbon steel.

Carbon and low-alloy steels offer good resistance to cleaning solvents, including inhibited acids at rather elevated temperatures. However, it must not be assumed that all carbon and low-alloy steels exhibit the same corrosion characteristics in inhibited acids (9, 10, 11).

Stainless steels are, in general, more resistant to corrosion than low-alloy steels. This particularly is true of the 18-8 types of chromium-nickel alloys. This is less true of the chromium stainless steels. Care should be exercised in the use of inhibited acid in contact with the latter.

Galvanized iron cannot be cleaned by either inhibited acids or strong alkalies. Neither should be used unless corrosion has already so nearly destroyed the galvanized coating that its complete removal is of no consequence.

Copper and Copper Alloys

Copper and copper alloys are, in general, resistant to nonoxidizing acids, alkalies, and organic solvents.

Nickel and Nickel Alloys

Nickel and nickel alloys are resistant to nonoxidizing acids, alkalies, and organic solvents.

Aluminum and Aluminum Alloys

Aluminum and aluminum alloys are becoming more extensively used in the fabrication of heat-exchange equipment. The solvent cleaning of such equipment must not be attempted with inhibited acid or with strong alkali. Strongly oxidizing acids and mild alkalies may be used.

Tin and Lead

Block tin is seldom encountered but is quite resistant to attack by the common solvents under the conditions employed. The same may be said of lead, which might be used for gaskets, calking, or special fittings. Further, tin-lead solders are not subject to serious attack.

Chemistry of Chemical Cleaning

Inorganic Deposits. In general, inorganic deposits are removed because the acid solvent reacts with the compounds in the deposit converting them to new compounds which are soluble in the aqueous solution. For example

Carbonates

$$CaCO_3 + 2HCl = CaCl_2 + H_2O + CO_2$$

The CO_2 is evolved as a gas and the $CaCl_2$ is readily soluble in water. Other carbonates behave in the same way.

Oxides

$$Fe_2O_3 + 6HCl = 2FeCl_3 + 3H_2O$$

This reaction goes on more slowly than the one used in the first illustration. However, the product ferric chloride is soluble in the acid medium.

Certain oxides present special problems: Cuprous oxide, because it is not readily soluble in low concentrations of hydrochlori acid, and manganese dioxide because, if it does dissolve in hydrochloric acid, it causes the liberation of free chlorine which ca create a serious corrosion hazard.

Sulphides. Iron sulphide dissolves readily but with evolutio of hydrogen sulphide, a very poisonous gas, which must b handled with proper safety precautions in mind. Copper su phides are not readily soluble. Zinc sulphide is intermediate i properties; it will dissolve readily under the proper conditions.

Calcium Sulphate. This compound, which is found much mor often in boilers using raw water than in heat-exchange equipment, is difficult to remove. For all practical purposes it is onl slightly soluble in hydrochloric acid. One method of treatmen which is effective in some cases is to use alternate alkaline and aci stages. Some calcium-sulphate deposits respond to treatmen with hot strong caustic solutions (5), but this method has no been widely used to date. The cost and danger of handling th solvent tend to discourage its use.

Silicates. The chemistry of the silicates is rather comple It will serve our purpose here to indicate that there are two general methods of removing difficult deposits:

1 The use of hydrochloric acid containing addition agen which increase the solubility of the silicates.

2 The use of alternate alkaline and acid stages.

Usually laboratory tests and previous experience will guide t engineer in deciding which procedure to use.

There are some other miscellaneous substances which are encountered infrequently, but the ones cited are of the most ge eral interest and will serve well to illustrate the problems i volved. One other may be mentioned only because it presen an unusual corrosion problem. Calcium sulphite will dissolve hydrochloric acid but when it does, it forms a strong reduci agent, sulphurous acid, which destroys many of the organic-ty inhibitors that are used in chemical cleaning. This may resu in serious acid attack on the metal unit.

Organic Deposits. As suggested, much of the task of removi organic materials involves physical changes rather than chemic reactions. Solution of the deposit in a suitable organic solve or dispersion in aqueous medium are the most common metho The chemical reactions involved include the following

(*a*) Saponification by strong alkali.

(*b*) Oxidation by alkaline or acid solutions containing stro oxidizing agents.

(*c*) Reaction with concentrated sulphuric acid.

These few chemical generalizations are sufficient for this di cussion.

Precautions

There are several other problems to be considered if chemic cleaning is to be done without danger to equipment and to pe sonnel.

For the safety of both equipment and personnel, it must recognized that explosions have occurred in a few instances whe proper precautions have been wanting. Hydrochloric acid no mally reacts with iron

$$Fe + 2HCl = FeCl_2 + H_2$$

Hydrogen is evolved. Substances known as inhibitors are us in the acid to inhibit or discourage this reaction, but the be inhibitors cannot stop it completely. If the hydrogen which evolved is allowed to become mixed with air in a confined spac a spark, a lighted match, cigarette, or a flame tool, can ignite t mixture and a serious explosion may result. It is essential th ydrogen and other gaseous reaction products be conducted hrough proper vent lines away from confined spaces and from ources of ignition. "No Smoking" and "No Welding" signs hould be posted where necessary.

Specifically, for the protection of personnel, it is important that those handling acids, alkalies, inhibitors, and other compounds sed in chemical-cleaning operations be instructed properly in methods of handling and be clothed properly with goggles, rubber uits, and rubber gloves when these are necessary.

Proper equipment should also include devices for detecting poisonous gases, which may be formed by chemical reaction between solvent and fouling deposit, and suitable gas masks or respirators, to be worn when needed.

The following are some examples of hazardous gases:

1 Hydrogen sulphide is extremely poisonous and is frequently ncountered in chemical cleaning work

$$FeS + 2HCl = FeCl_2 + H_2S$$

2 Carbon dioxide is evolved from all deposits containing calcium carbonate when inhibited acid is applied. It is not poisonous and is not normally considered dangerous. It can cause suffocation. The gas is much heavier than air and tends to settle into low places.

3 Sulphur dioxide is formed from deposits containing sulphites. It is extremely disagreeable and severely irritating to respiratory organs.

For the protection of metal equipment there are some further considerations. It is not enough to know how each metal behaves in inhibited acid alone. One must consider also the possible effect of certain constituents in the deposit on the relation between solvent and metal. Oxidizing agents such as ferric-ion compounds (6), copper compounds, manganese dioxide, and chromates, tend to cause the corrosion rate to be higher than it would be otherwise. Special tests may be necessary to determine whether this effect will be serious. Reducing agents such as sulphites also have an adverse effect on many inhibitors. These facts must be properly anticipated and investigated if chemical cleaning is to be done safely.

Methods of Application

Chemical cleaning is generally performed by handling the solvent or solvents in tank trucks equipped with means of diluting concentrated aqueous solvents with water to the desired percentage concentration, of heating with live steam to the desired temperature, and pumping the solvent at the proper strength and temperature to the unit to be cleaned. This requires that steam and water lines be run to the truck location and a solvent line from the truck to the unit. Circulating the solvent during the reaction period may or may not be necessary or desirable. If it is, a return line from the unit to the pump truck becomes necessary. The need for proper venting has already been emphasized. A typical hookup is shown in Fig. 1.

The proper reaction time between solvent and deposit is determined by chemical tests and by the experienced judgment of the chemical-cleaning engineer. The solvent must then be disposed of, the unit flushed thoroughly with water and, following the use of acid solvents, a neutralizing wash applied.

If the solvent has dissolved most of the deposit and caused the emainder to slough away from the metal surface so it can be vashed out, the chemical cleaning would be considered successful. A nearly pure calcium carbonate or phosphate deposit will lissolve practically completely in a short time, but an iron-oxide leposit, for example, dissolves slowly, while much of the deposit ends to slough away. It would be impractical in most cases to ontinue the treatment to the point of dissolving 100 per cent of the deposit. In general, then, it is necessary to anticipate the presence of sloughed material. In many cases normal flow of water through a heat exchanger will remove this sloughed material. In other instances it is necessary to provide some means of washing or scraping out this material. This is one of the most serious difficulties involved in removing certain deposits from the shell sides of tube and shell exchangers without dismantling the units. Fortunately, in most other types of equipment it is not a serious problem.

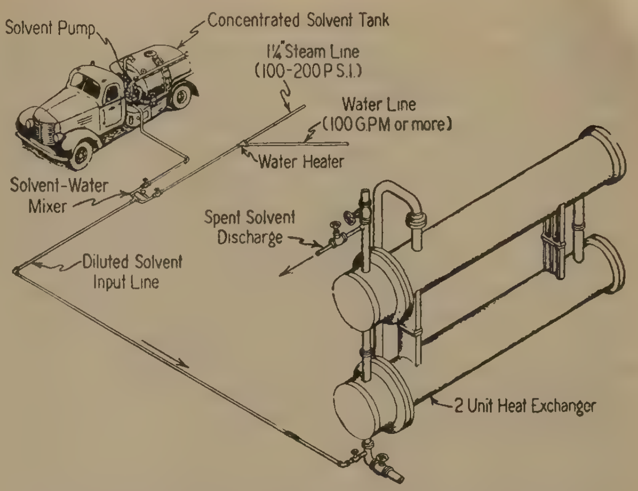
Solvent Pump
Concentrated Solvent Tank
1¼" Steam Line (100-200 P.S.I.)
Water Line (100 GPM or more)
Water Heater
Solvent-Water Mixer
Spent Solvent Discharge
Diluted Solvent Input Line
2 Unit Heat Exchanger

Fig. 1 Typical Hookup for Chemical Cleaning of Heat-Exchange Equipment

The properties of metals encountered in chemical cleaning are given in Table 1.

Case Histories

Power Plant (Turbine Condensers). The degree of fouling which can be permitted before cleaning is necessary depends upon the individual circumstances (7). In a large utility generating station every tenth of an inch of vacuum lost because of fouling is important. In the powerhouse of a small manufacturing company, generating part or all of its own power, the loss of vacuum may become several inches before the operator is seriously concerned.

Most generating-station turbine condensers are chemically cleaned on the water side or steam side or both by filling with dilute hydrochloric acid, containing inhibitors and wetting agents, allowing a soaking period of 1 to 4 hr, and then flushing. The water side is easily flushed with regular circulating water. The gain in vacuum or drop in back pressure probably will be somewhere between 0.1 in. and 1.5 in.

On the other hand, a small condenser in a manufacturing plant may require two or more fillings of inhibited hydrochloric acid to restore several inches of lost vacuum. Perhaps it would be better economy to have cleaned the condenser at more frequent intervals using correspondingly less solvent and time and having a much higher average efficiency of operation of the unit.

If a serious slime condition exists in the cooling-water side of a turbine condenser in addition to scale deposits, it may be necessary to remove the slime by chlorine, copper sulphate, or hot alkaline pretreatment and then to use inhibited acid to remove the scale.

Feedwater Heaters. High-pressure-stage feedwater heaters present a problem in chemical cleaning. Low-pressure heaters are usually fouled with deposits containing copper and iron oxides. Dilute inhibited hydrochloric acid will generally remove them easily.

In high-pressure-stage heaters the same deposits are encountered and have the same physical appearance. However, experience has shown that they cannot be removed with straight inhibited hydrochloric acid. The reason for this is that they generally contain decomposed oil in amounts of from approximately 2 to 30 per cent. This oil, even in the smaller quantities, interferes with the solution of the deposits in acid.

An example is a Foster Wheeler four-pass horizontal heater containing 760, 14-gage Admiralty, 0.75-in-OD tubes 18 ft long with 2690 sq ft of heating surface. The unit was a fourth-stage heater operating at approximately 400 F, deposits consisting of a very thin scale containing copper oxide and hydrated iron oxide. Inhibited hydrochloric acid containing a surface-tension-lowering agent was added to the heater at 155 F and slowly pumped through the unit, expending the used solvent to the sewer. The unit on casual observation looked clean; however, later operating data indicated that it had not been cleaned to any appreciable extent.

The deposits were reanalyzed and an infrared analysis showed that there was approximately 2.2 per cent oil present. Samples from a fourth-stage heater in another unit, operating under the same conditions, were obtained. These had the same analysis, and an infrared analysis indicated that there was approximately 2.2 per cent oil present. An oxidizing solution was pumped through the second heater and circulation was continued for 1 hr, following which the solution was displaced to the sewer with water. Inhibited hydrochloric acid was then added. This time the heater was clean.

The same types of deposits in high-pressure feedwater heaters have been encountered in several power plants. Oftentimes one stage of the oxidizing agent followed by an acid solvent is not sufficient. On some heaters as many as four stages of each type of solvent have been necessary.

Some interesting performance data on feedwater heaters chemically cleaned are contained in Table 2.

Steel Mill Coke Plan's (*Wash-Oil Coolers*). For the recovery of light oils from coke-oven gas, the gas is scrubbed with a high-boiling petroleum oil called wash oil. The wash oil picks up light oils in a light-oil scrubber, goes to the light-oil recovery plant, where the light oils are distilled out, then back to the scrubber. On the way back, it goes through wash-oil coolers which cool it with water to the appropriate temperature for efficient pickup of light-oil vapors.

The coolers consist of cast-iron sections arranged in columns and rows with a system of water spray nozzles overhead and catch basins underneath. The outer surfaces of the cast-iron sections were covered with iron oxides formed as a result of corrosion of the cast iron. Between the outer corrosion products and good metal underneath, was the transition layer of partly corroded metal which is typical of graphitic corrosion on cast iron. This layer tends to discourage the sloughing away of the corrosion products leaving a clean metal surface.

The procedure for cleaning was to spray warm inhibited acid over the coolers through the regular spray system while the stands were kept warm by allowing some oil circulation through them. After about 10 min, the acid spray was stopped, the stands allowed to soak a short time, and then were washed down with high-pressure water hoses to help slough away the heavy iron-oxide layers. This cycle was repeated until the coolers were sufficiently clean. Only part of the coolers could be treated at one time while the others were in use. From 3 to 5 hr were required for each part.

After cleaning, the temperature of the oil leaving the coolers was 13 to 14 deg F lower than before cleaning, with the same rates of water and oil flow to the coolers.

Refinery Equipment. Few industries use more heat-exchange equipment, have more trouble with fouling, or more interesting chemical-cleaning problems than the petroleum industry. The deposits found in cooling-water systems do not present an unusual or peculiar problem, but the product systems present many problems. The deposits are essentially sulphides of the heat-exchanger metals, and organic matter in various states of decomposition and polymerization. Iron sulphides are soluble in inhibited acid, but the H_2S evolved is a very poisonous gas. The copper sulphides and polymerized organic matter are not readily soluble. When the latter occur on the shell side of a tube and shell exchanger the problem is aggravated by the possibility of the spaces between the tubes being plugged solid so the solvent cannot come in contact with the deposit, and by the difficulty of removing material which sloughs loose but does not dissolve.

Heat Exchangers. Following is described a case in which a mixed organic-inorganic deposit was found in the tubes of a horizontal tube and shell exchanger. A desalted Illinois crude oil passes through the tube side; in at approximately 275 F and out at 310–320 F. An attempt was made to clean the tube side with alternated stages of detergent solution and acid, but without success.

Tests conducted in the laboratory on the deposit showed it to be a mixed organic-inorganic material. The results of the tests may be shown as follows:

Organic matter soluble in carbon tetrachloride, per cent.....	31
Organic matter insoluble in carbon tetrachloride, per cent.	8
Inorganic matter soluble in hydrochloric acid, per cent......	49
Inorganic matter insoluble in hydrochloric acid, per cent....	12
The total soluble in hydrochloric acid and carbon tetrachloride, per cent..............................	80

TABLE 2 TERMINAL TEMPERATURE DIFFERENCES OF TURBOGENERATOR FEEDWATER HEATERS BEFORE AND AFTER CHEMICAL CLEANING

	Unit 8		Unit 9		Unit 10		Unit 12	
	Before	After	Before	After	Before	After	Before	After
4th-Stage heater:								
Actual steam temp, deg F...	646.9	629.6	(Not taken)	620.5	669.5	633.4	574.2	562.7
Avg steam press, psia.......	209.9	200.0	221.2	214.1	219.9	221.5	152.1	148.7
Temp water leaving htr, deg F	378.8	380.0	388.1	387.4	386.5	390.2	352.7	353.8
Terminal temp diff,[a] deg F..	+7.0	+1.8	+2.2	+0.1	+3.0	+0.2	+6.9	+4.0
8th-Stage heater:								
Actual steam temp, deg F...	504.3	486.7	(Not taken)	471.3	513.5	499.7	394.2	371.4
Avg steam press, psia.......	96.2	91.9	100.4	97.1	101.0	101.5	47.7	44.6
Temp water leaving htr, deg F	311.4	316.8	318.3	324.2	313.0	324.2	269.7	267.4
Terminal temp diff,[a] deg F..	+13.5	+4.9	+9.8	+1.5	+15.5	+4.7	+7.8	+6.5
11th-Stage heater:								
Actual steam temp, deg F...	314.2	291.8	(Not taken)	315.6	335.9	309.8	180.2	177.8
Avg steam press psia........	39.3	36.0	40.2	38.1	40.8	39.4	7.9	7.7
Temp water leaving htr, deg F	259.3	259.1	259.3	263.9	254.0	265.0	173.2	170.3
Terminal temp diff,[a] deg F..	+6.9	+1.8	+8.3	+1.1	+14.5	+1.3	+9.1	+10.8
14th-Stage heater:								
Actual steam temp, deg F...	183.6	182.5	(Not taken)	184.4	201.8	186.3	(Not taken)	
Avg steam press, psia.......	8.8	8.2	8.4	8.2	9.2	9.1	(Not taken)	
Temp water leaving htr, deg F	178.2	176.9	171.7	176.2	167.8	180.0	(Not taken)	
Terminal temp diff,[a] deg F..	+9.0	+7.0	+13.4	+7.8	+21.4	+8.9	(Not taken)	

[a] Based upon saturated steam temperature.

TABLE 3 DESIGN DATA ON HEATERS IN TABLE 2

	Unit 8	Units 9 and 10	Unit 12
4th-Stage heater:			
Manufacturer	Foster-Wheeler	Foster-Wheeler	Foster-Wheeler
Total area, sq ft	1340	1340	2830
Nominal over-all heat transfer, Btu per hr	1924×10^6	20.65×10^6	59.03×10
Turbine-throttle steam flow, lb per hr	280000	280000	570500
8th-Stage heater:			
Manufacturer	Foster-Wheeler	Foster-Wheeler	Foster-Wheeler
Total area, sq ft	1340	1340	2820
Nominal over-all heat transfer, Btu per hr	18.31×10^6	18.90×10^6	41.65×10^6
Turbine-throttle steam flow, lb per hr	280000	280000	570500
11th-Stage heater:			
Manufacturer	Foster-Wheeler	Foster-Wheeler	Foster-Wheeler
Total area, sq ft	1363	1363	2100
Nominal over-all heat transfer, Btu per hr	21.36×10^6	22.80×10^6	44.58×10^6
Turbine-throttle steam flow, lb per hr	280000	280000	570500
14th-Stage heater:			
Manufacturer	Foster-Wheeler	Foster-Wheeler	...
Total area, sq ft	1025	1025	...
Nominal over-all heat transfer, Btu per hr	21.42×10^6	20.68×10^6	...
Turbine-throttle steam flow, lb per hr	280000	280000	...

The method of cleaning decided upon was to use an emulsified solvent containing both inhibited hydrochloric acid and organic solvent. Two batches of emulsified solvent were circulated through the unit, then it was flushed, neutralized, and put back in service for another complete run.

It is interesting that, in the same system, in the exchanger folowing the one just described, where crude-oil temperature rises to 400 F, the deposit was quite different:

Organic matter soluble in carbon tetrachloride, per cent	30
Organic matter insoluble in carbon tetrachloride, per cent	32
Inorganic matter soluble in hydrochloric acid, per cent	17
Inorganic matter insoluble in hydrochloric acid, per cent	21
Total soluble in hydrochloric acid and carbon tetrachloride, per cent	47

Tubes taken from this unit were treated with the emulsified solvent in the laboratory by circulating through a single tube with a laboratory pump. The tube did not clean up. There was not sufficient material soluble in the solvent to permit the insolubles to slough away. This is pointed out merely to illustrate that a method which may be quite successful on one unit may fail on another operating under different conditions.

Gasoline Condensers. In a Texas refinery a heavy calcium-carbonate scale was removed from the cooling-water side of gasoline condensers with inhibited hydrochloric acid. The results reported are interesting: After chemical cleaning, the gasoline leaves the condensers 20 deg F cooler than when the units are mechanically cleaned. The operator reported that this 20 deg F lower temperature will reduce evaporation losses to the extent that the cost of chemical cleaning will pay out in less than 2 weeks of operation.

Marine Lubricating-Oil Coolers. The lubricating-oil coolers on many ships need frequent cleaning to remove from the oil side a heavy greaselike material which is primarily hydrocarbons with some sulphur containing organic matter (8).

The usual procedure is to use an emulsion, consisting of an organic solvent emulsified with an aqueous alkaline solvent. The solvent is used at elevated temperatures and from 2 to 4 hr contact time are usually sufficient. Occasionally it is necessary to follow the emulsion with an acid stage if corrosion products have been formed in or carried into the coolers.

Synthetic-Rubber Plant. An interesting operation conducted during the war was the chemical cleaning of all the heat-exchange equipment in a synthetic-rubber plant. The plant was designed with a rated capacity of 90,000 tons of synthetic rubber per year. It was laid out in three identical production lines designated A, B, and C. Each line consisted of 24 jacketed reactors plus supply, circulating and overflow lines, with a total estimated volume of 16,000 gal; circulating-water system with two 3-stage condensers, two booster condensers, with supply and return lines with an estimated total volume of 4700 gal; refrigerated water system with two aftercoolers on the 3-stage condensers, two aftercoolers on the booster condensers, two butadiene condensers (shell side) and two other heat exchangers, with supply and return lines, total volume estimated as 3000 gal.

The deposits were essentially iron oxides and were removed with dilute inhibited hydrochloric acid. Approximately one 8000-gal tank car of concentrated hydrochloric acid was required on each line. Three tank cars were used in the entire cleaning operation. It will not be necessary to describe the details of the work as this illustration is intended to show the large amount of cleaning that can be done in a relatively short time. The first line was cleaned between 8:00 a.m. Monday and 8:00 a.m. Tuesday. It was put back into operation. The second line was cleaned on Wednesday, and the third line was cleaned on Friday. Each line was put back into full production in less than 48 hr.

Heating System. In an automobile plant the hot water for heating the plant and office building is supplied by using exhaust steam from the forges in Alberger heaters. The heaters are vertical, two-pass, tube and shell exchangers, designed for a heat transfer of 60,000,000 Btu per hr at 3 psi saturated-steam pressure and a water flow of 4000 gpm. The original copper tubes had been replaced by Admiralty-metal tubes. It was estimated that this would reduce the normal heat-transfer rate from 60,000,000 to 46,000,000 Btu per hr.

Oil and grease from the forges had gotten past the oil separator, and the shell side (steam side) of the heaters was fouled until grease stood 6 to 8 in. deep on the lower tube sheet. The tube side was fouled with a readily soluble water-deposited scale.

The treatment consisted of using an organic solvent to remove the 6 to 8-in. layer of grease on the tube side, then an alkaline detergent to remove the oil film from the entire shell side, finally an inhibited acid solution to remove the water scale from the tube side. The acid solution was circulated through the unit first in one direction, then in the opposite direction. The whole operation was completed on each heater in 6 hr. Tubes later taken from the unit were thoroughly clean inside and out. Performance data on both heaters are given in Table 4.

TABLE 4 PERFORMANCE DATA ON AUTOMOBILE-PLANT HEAT EXCHANGERS

Heater no. 1	Outside temp, deg F	Temp of water to heater, deg F	Temp of water leaving heater, deg F	Steam press on shell side, psi
Jan. 15	31	145	160	2.5
Jan. 17	Cleaned unit			
Jan. 20	36	173	191	1.6
March 13	15	136	154	1.4
Heater no. 2				
Jan. 15	31	145	160	5.3
Feb. 18	Cleaned unit			
March 4	5	146	176	1.3

Unfortunately, data on rates of flow of water through the units are not available. Before cleaning it was necessary to operate both units continuously; since cleaning it has not been necessary to operate more than one heater at any time. Also, the supply of steam from the boilers, which is used when the forges are not operating, has been reduced materially.

Salt Manufacture. Multiple-effect evaporators are used in the salt industry to concentrate the salt brine. The chemical cleaning of these evaporators is well illustrated by experiences in a plant in Michigan. The deposits in the first effect are almost pure calcium sulphate. Those in the second effect are mixtures of calcium sulphate and calcium carbonate. The deposits in the third and fourth effects are almost pure calcium carbonate. Manual cleaning generally requires a crew of 8 to 12 men to clean one set of pans in approximately 8 days. In chemical cleaning, inhibited hydrochloric acid is added, filling to slightly above the upper tube sheet. Where heat is required, steam is vented into the solvent. Within 4 to 6 hr the third and fourth effects are free of scale, the second effect is almost clean, but the first effect still contains considerable calcium sulphate scale. The tubes in the first and second effect are then turbined in 6 hr with a two-man crew. Salt production is increased several tons per day following the cleaning.

Cost Considerations

One consideration which has not been discussed and which is difficult to approach satisfactorily in a general way, is the question of the cost of chemical cleaning as compared to the cost of other methods of cleaning heat-exchange equipment. Many factors enter into comparative costs which vary tremendously from one situation to another. The cost per unit depends upon how many units can be cleaned simultaneously by the same crew of cleaning engineers, and the amount and type of scale present. Another factor of cost to the operator is the expense of preparing the unit for cleaning. Here, chemical cleaning may offer considerable advantage by eliminating the need for building scaffolds, dismantling the unit, and so on. Still another very important consideration is the saving of time, if down time means the lowering of production schedules. In many instances this factor alone pays for the cost of chemical cleaning many times over. In other instances it does not represent a saving.

Cleaning Schedules

When heat-exchange equipment is to be dismantled and cleaned manually, it is natural to let it go as long as possible even though its inefficient operation is costing dollars every day of operation. If the unit is going to be dismantled and cleaned manually, the time and cost are about the same whether the unit has $^1/_{64}$ in. of deposit on the metal surfaces or 4 times that much. So the job is put off as long as possible.

Suppose the deposit is a calcium-carbonate scale accumulating at the rate of $^1/_{64}$ in. every 6 months. The deposit will be $^1/_{16}$ in. at the end of 2 years. If the unit is to be cleaned chemically, the amount of solvent required is directly proportional to the amount of scale, so whether $^1/_{64}$ in. is removed every 6 months during the two years, or $^4/_{64}$ in. at the end of the 2 years, the amount of solvent used would be approximately the same. But by applying one quarter of the solvent every 6 months, the overall average efficiency of the unit is much greater than it would be if the deposit accumulates for 2 years. There are, to be sure, other costs for chemical cleaning than the cost of chemicals, but that is usually the largest single item.

With chemical-cleaning programs, it is well to consider the advantages of cleaning at intervals such that the cleaning can be done quickly and easily, and the unit kept at high average operating efficiency. When units are allowed to become badly fouled, tubes may become plugged and spaces between the tubes become packed solid so solvent cannot get to the deposit. The cost of chemical cleaning may appear excessive and the results may be unsatisfactory. Then if the operator has to dismantle the unit and rod-out plugged tubes and finish the cleaning job, he is quite likely to feel that someone has misrepresented the advantages of chemical cleaning.

Conclusion

It is hoped that this discussion has covered the primary applications of chemical cleaning to heat-exchange equipment. This highly specialized field of chemical engineering is not a cure-all for the maintenance department's headaches, but with chemical-cleaning engineers working in co-operation with maintenance men, operating engineers, water-treatment engineers, and those who design and install heat-exchange equipment, many of the existing problems can be solved. It is not a field for patent-medicine methods. Rather, it is a challenge to well-trained and competent chemists, engineers, and technical-service personnel. In its proper applications, it represents a new and better way of doing a very old job.

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Scale Formation and Control in Compression Distillation of Sea Water

By J. J. CAMPOBASSO[1] and ALLEN LATHAM, JR.[2]

Various principles which have been used to combat the formation of scale in sea-water distillation equipment are outlined briefly. Data on the rate of scale formation in the compression type of distillation unit are presented. Results obtained by periodic chemical cleaning and low-temperature operation are discussed.

OPERATORS of distillation equipment for production of fresh water from sea water have been plagued by fouling of the heat-transfer surfaces through formation of scale ever since the earliest water stills were placed in service. Solutions to this problem have been made by use of several distinctly different principles, or combinations of them. These principles are as follows:

1 Periodic mechanical scale removal.
2 Periodic chemical scale removal.
3 Periodic scale removal by thermal shock which causes differential expansion between the scale and the metallic heat-transfer surfaces.
4 Periodic scale removal by temperature change, which induces bending of the metallic heat-transfer surfaces.
5 Periodic scale removal by pressure variation, which induces bending of metallic heat-transfer surfaces.
6 Continuous scale prevention by vacuum operation at low temperatures, which completely alters the scale-forming tendencies in sea water.
7 Continuous scale prevention by feedwater treatment.

The scale-forming tendencies of sea water vary markedly in different localities, particularly near the shore, and the chemical constitution of the impurities in sea water is so complex that there is often a marked variation in the chemical analysis of the scale from different points within distillation equipment. Consequently it is hazardous to predict the exact results which will be obtained by application of any of the foregoing principles. However, practically all of them have been used successfully at one time or another.

Data Derived From Scale-Removal Applications

The object of this paper is to present data on the results obtained through the application of some of these principles to the removal or prevention of scale in the compression type of distillation equipment. This form of still was widely used during the war for military water supply and is now in use in several marine and industrial installations. All of the data reported below were taken on compression stills which used a short vertical tube, natural circulation, callandria type of condenser-evaporator with boiling inside the tubes.

These units were designed for periodic mechanical removal of scale with conventional hand or power-operated tube cleaners, and the general practice was to use this method of scale removal. Consequently the broadest background of experience has been with mechanically cleaned units. Under typical conditions, a total operating time of 720 hr was obtained before it became desirable to shut down for cleaning. The characteristic relationship between cumulative output of distilled water and scale coefficient is shown by curve A in Fig. 1. The labor required for mechanical cleaning averaged about 0.4 man-hour per 1000 gal of distilled water.

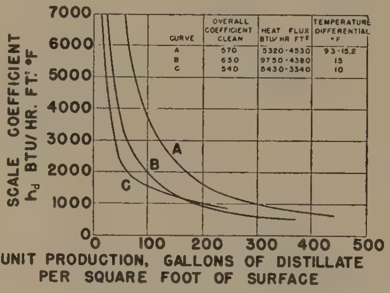

Fig. 1

Curve A in Fig. 1 is for a typical run on water from the seacoast of Virginia where the sea water contained very little contamination from inland water. Curves B and C indicate the results of special runs on different sizes of the same type of unit operating on water from Boston harbor. The differences between B and C may be attributed to the fact that unit B was run at 15 deg F temperature differential, as compared with 10 deg F for unit C. The disparity between curves A and B-C is probably partly due to the difference between contaminated harbor water and sea water and partly due to the difference in tube length, as shown in the following table:

Unit	OD in.	Tube wall, in.	Length, in.
A	$^3/_4$	0.049	$22^1/_4$
B	$^3/_4$	0.049	$16^1/_4$
C	$^3/_4$	0.049	$16^1/_4$

In all three cases the temperature of the boiling sea water was approximately 213 F. All of these units had been used prior to the test and were carefully cleaned and inspected before each test. For this purpose, chemical cleaning, supplemented by mechanical cleaning, was used.

These curves are all based upon operation of complete full-scale distillation units on which it was not feasible to take such measurements as tube-wall temperatures. Over-all temperature differential and distillate production were measured and used for calculation of over-all heat-transfer coefficients. The resistance due to scale formation was assumed to account for the entire difference in over-all resistances calculated for the condenser-evaporator in the fouled and clean conditions, according to the relationship

$$\frac{1}{U} = \frac{1}{U_c} + \frac{1}{h_d}$$

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where

U = over-all coefficient
U_c = over-all coefficient with clean tubes
h_d = scale coefficient

No attempt was made to adjust U_c for variations in temperature differential and heat flux.

Several different methods of chemical treatment, both continuous and periodic, were tried with varying degrees of success. One of the most reliable methods found was periodic cleaning with sodium acid sulphate. A typical procedure for use of this chemical is to circulate a solution of about 2 lb $NaHSO_4$ per gal of sea water for a period of about $1^1/_2$ hr after each 300 hr of normal operation. Under some conditions it has been found desirable to flush out the spent solution and run a second $1^1/_2$-hr treatment with fresh solution. This may be carried out without even cooling the equipment below operating temperature and no lost time is involved in restarting the unit. By this method, only about 1 per cent of the time is devoted to scale removal. Runs of several thousand hours' duration have been made without shutdown for mechanical cleaning. Inhibited muriatic acid can be used in place of sodium acid sulphate to accomplish substantially the same results. Average consumption of chemicals is 1.5 lb of $NaHSO_4$, or 0.5 lb of HCl per 1000 gal of distillate. It has been found advantageous to conduct this periodic cleaning when the degree of fouling corresponds to a scale coefficient of about 1700.

Low-Temperature Operation

One of the most effective ways to prevent scale formation in sea-water distillation equipment appears to be operation at low temperature. A trial unit was operated on sea water from Boston harbor for a period of about 1200 hr, with practically no evidence of scale formation. As compared with normal operation at a boiling temperature of about 213 F, this run was made under partial vacuum to give a boiling temperature of 150 F. Although the amount of experience available on this method of scale prevention in compression stills is meager, hearsay evidence on corresponding experience with other types of sea-water stills tends to confirm the foregoing indication that when the boiling temperature is held down to 150 F, scale formation is prevented. Although low-temperature operation appears to eliminate scale formation, it permits a film of slime to form, giving a scale coefficient of about 1300, which remains constant.

Conclusions

1 The rate of scale formation in current models of compression sea-water-distillation units is such that 200 to 300 gal of distillate can be produced per square foot of evaporation surface before it is necessary to clean the surface.

2 Periodic treatment with solutions of sodium acid sulphate or hydrochloric acid can be used to obviate the need of mechanical cleaning.

3 Vacuum operation to give boiling temperatures of 150 F offers a good possibility for prevention of any significant formation of scale.

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Cleaning Tubular Heat Exchangers

By P. F. DOUGHERTY[1] and C. H. BROOKS,[1] PHILADELPHIA, PA.

A discussion is presented covering various methods available for cleaning tubular heat-transfer equipment, with particular emphasis on the use of an emulsifying oil for "in-place" cleaning, and on the use of sandblasting. The relative merits of the in-place and the mechanical methods are outlined as well as the factors to be considered in selecting a particular procedure for a given application.

INTRODUCTION

THE cleaning methods for tubular heat-transfer equipment fall into two general classes. One class consists of those methods which perform the cleaning without dismantling the equipment, by means of the circulation of a solution which chemically and/or physically removes the fouling material. The other class includes all those methods where the equipment is partially or wholly dismantled, and the fouling agent removed by a purely mechanical means.

The adaptability of the in-place methods must be determined by physical and chemical examination of the fouling substance. Laboratory solubility tests must also be made of the material. A plant scale test under close supervision should then be made, if it be found in the laboratory that a significant portion of the fouling material be soluble.

If no suitable method is found for cleaning in place, one of the various mechanical methods must be used. The selection here will be determined by the nature of the deposit and extent of the deposit, as well as other factors which will be discussed later in the paper.

It is obvious that the efficiency of cleaning and the frequency with which it must be performed, can be determined only by periodic heat-transfer surveys. In many cases, however, it should be possible to establish certain criteria which may be used by the operators, without the necessity of performing a complete survey

CLEANING IN PLACE

In order that equipment may be cleaned in place the fouling material must be capable of removal by one of the following methods:

1 Chemical reaction with, and subsequent solution in, an acidic or alkaline solution.

2 Solution of a portion of the fouling material, i.e., a binder, in an aqueous cleaning fluid, thus permitting the undissolved portion to be flushed out. The cleaning fluid may be hot or cold water or acidic or alkaline solutions. The flushing agent may be the cleaning fluid used, or another liquid, or emulsifying oil.

3 Suspension of the fouling agent in the cleaning fluid.

4 Complete solution of the fouling agent or of the binder in an organic solvent.

5 Combinations of the foregoing methods.

Our experience indicates that cleaning in place has the following advantages:

1 Down time for equipment is reduced.

2 Dismantling of equipment with resultant damage and high labor cost, is avoided. In those cases where tube bundles cannot be removed, no other methods are available for cleaning the shell side.

3 Cleaning is more thorough. The solution will clean parts which would be difficult or impossible to reach with cleaning tools.

4 Cleaning is more uniform. There are no small particles of scale remaining on the surface. On the contrary, mechanical cleaning usually leaves small particles of scale on the surface, which act as nuclei for the rapid formation of new scale.

5 There is practically no loss of metal.

6 The metal surface is left smooth in contrast to mechanical cleaning which frequently results in scoring which provides a place for lodgment of fouling material.

As just pointed out, laboratory tests should be made to determine the feasibility of the use of a particular in-place method. This work should be followed by closely supervised plant scale tests to confirm the laboratory results and to develop the actual procedure to be followed, if the method be adopted.

Our experience has indicated that the normal procedure of steaming the equipment for extended periods prior to cleaning should not, as a rule, be followed when cleaning in place is planned. In one case it was found that the tubes appeared almost clean when the equipment was opened without steaming, whereas they were almost completely plugged with a rubbery mass after steaming for several hours. Prolonged steaming, in most instances, results in the removal of the more volatile components of the fouling material and the production of a residuum which is far less soluble than the original fouling substances.

Prior to the adoption of an in-place method of cleaning a particular piece of tubular equipment, investigation should be made to determine the following:

(*a*) Whether there be evidence of severe corrosion or deep pitting. The removal of the scale may uncover actual perforations in the tubes, which are plugged with scale.

(*b*) Whether, in the course of normal service, any portion of the tubes becomes plugged completely. If such be the case, the equipment must be opened and the tubes unplugged prior to in-place cleaning. Otherwise, the solution will be unable to contact the fouling material and the equipment will not be thoroughly cleaned.

Solution Cleaning. The use of inhibited acid has been described and discussed at length by others. We have found that the use of this technique is most advantageous on the water side of coolers. It should be noted that, in at least one instance the fouling material was only partly soluble in the acid. The solution of this portion, apparently acting as a binder, permitted the remainder to be flushed out by the circulation of the acidic solution.

Solution of Scale Binder. The basis of this method of cleaning is the solution of sufficient of the fouling material to loosen the remainder, at the same time that a sufficiently high circulation rate is maintained so that the loosened material is mechanically carried out of the equipment. In many cases it is advantageous

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[1] Engineering Division, Manufacturing Department, Sun Oil Company.

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NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society. Paper No. 48—A-116.

to intersperse a boiling of the solution, by steam applied to the other side of the tubes, as well as to reverse the flow of circulating medium.

An alkaline solution was used to clean certain exchangers in a solvent refining unit, with the results shown in Table 1. An abnormally high circulation rate is believed to have assisted the cleaning in this case. In addition, no steaming was used prior to cleaning.

TABLE 1 RESULTS OF EXCHANGER CLEANING TESTS

Example	Solvent refining unit exchangers A	Gas oil-crude exchangers B	Absorption oil exchangers C	Gasoline feed-bottoms exchangers D
Cold end temp. difference, deg F:				
Before cleaning.......	185	..	46	40
After cleaning.........	146	..	2	12
Design...............	..	..	5	30
Hot end temp. difference, deg F:				
Before cleaning.......	155	..	115	91
After cleaning.........	94	..	91	75
Design...............	..	..	38	35
Over-all coefficient, Btu/h12/ft^2/deg F:				
Before cleaning.......	13.5	16	7.2	36.4
After cleaning.........	37.0	46	30.2	51.1
Design...............	42.5	..	35	60
Pressure drop, psi:				
Before cleaning.......	..	..	..	90
After cleaning.........	..	..	..	20

Another example of the efficacy of dissolving a binder from the fouling material is given by a set of "gas oil-crude" exchangers in a catalytic cracking unit, which was fouling rapidly on the crude side. Steaming-out had proved insufficient to remove the fouling material, which was approximately 70 per cent salt, with balance insoluble organic and inorganic material. Circulation of boiling water at a very low rate produced the results shown in Table 1.

Cleaning With Emulsifying Oil.[2] The use of an emulsifying oil, available in our refinery, is applicable where the fouling material is (*a*) completely soluble in the oil, or (*b*) can be held in suspension in the oil, and thus removed from the equipment.

This method was primarily developed for the cleaning of the oil (shell) side of absorption oil coolers. The fouling material was an emulsion with a greaselike consistency, composed of water, oil, resins, and inorganic solids. The use of heat or solvents broke this emulsion, allowing the resins and inorganic solids to settle out behind the baffles where they could not be removed by flushing with water or high or low-viscosity oils. Experience indicated that the presence of these solids accelerated the formation of the emulsion after cleaning. Hence a method to remove both the emulsion and the solids was required. Experiments indicated that mixing concentrated emulsifying oil, i.e., without dilution with water, with the fouling emulsion and then heating to obtain the desired fluidity, would permit these objectives to be attained.

After isolation of the equipment, the oil (shell) side was filled with emulsifying oil. The oil was heated to approximately 150 F and circulated until there was no change in the appearance of material entering the circulating tank. Circulation was then stopped, and the contents of the shell heated until a pressure of 50 psig was reached. The contents of the shell were then blown through the drain valve into the circulating tank. After washing with water, the unit was drained and returned to service. The improvement is indicated in Table 1.

Substantially the same procedure was followed in cleaning the feed-bottoms exchangers on a light gasoline-distillation tower, with the result shown in Table 1.

MECHANICAL CLEANING

When the various in-place methods prove to be inadequat resort must be made to the mechanical methods which inclu the following:

(*a*) Lancing with air, steam, or water.

(*b*) Drilling or "rodding" the obstructive material from th tubes.

(*c*) Sandblasting.

All of these methods require that the exchanger be at lea partially dismantled for cleaning, as distinct from the in-pla methods, where any dismantling is only for inspection purpose

The selection of the method depends upon the nature of th fouling material. It should be noted that it is decidedly adva tageous to steam both the tube and shell sides for as long as tin will permit, prior to mechanical cleaning. This steaming remov the tarry or oily components of the fouling material, thus i creasing the effectiveness of a given cleaning method.

The discussion which follows is confined to cleaning the "tube side of tubular heat exchangers.

Lancing. This general method employs a relatively sma high-velocity jet of steam or air and occasionally water. It applicable only when: (*a*) a low percentage of the tubes are com pletely plugged; (*b*) the fouling material is not tightly bound t the tubes; and (*c*) the fouling agent is not of a gummy or tarr nature.

Drilling and Rodding. Drilling and rodding consist of forcin a closely fitting rotating drill or a closely fitting rod through th tube. These methods are used when: (*a*) a high percentage of th tubes are completely plugged, (*b*) preliminary steaming leav considerable tarry or gummy material in the tubes, or (*c*) if th fouling material is tightly bound to the tubes (drilling only).

These two methods have certain serious disadvantages whic are common to any method, e.g., wire-brushing, which in essen scrapes the fouling material from the surface. These disadvan tages include the following:

(*a*) Excessive down-time and labor costs.

(*b*) Frequently a necessity for removing the tube bundle b fore it can be cleaned.

(*c*) Metal loss as a result of cleaning is excessive and/or un evenly distributed. Scoring of the tubes, which sometimes occur seems to accelerate the fouling rate.

(*d*) Small amounts of the fouling agent are left on the tub surface, which act as nuclei for the rapid build-up of the foulin agent.

Sandblasting. Sandblasting can be used to clean the inside o either steel or admiralty tubes when the fouling material is no removable by any of the previously discussed methods. Suc fouling material, in most cases, is composed of mixtures of tar resins, inorganic solids, and carbon. Its physical and chemic nature is such that an insufficient amount will dissolve in any o the solvents or solutions previously outlined to permit the sca to be removed.

The procedure which has been followed successfully, with n measurable metal loss, is as follows:

(*a*) After isolation and draining of the exchanger, it is steame for at least 6 hr, and longer if possible, to vaporize, or solidify an oily, tarry, or gummy materials. Otherwise, the sand will ten to stick to the surface, thus reducing its effectiveness.

(*b*) The exchanger is opened and any plugged or partiall plugged tubes are opened by lancing, drilling, or rodding. To us the sandblast method effectively, an opening through the enti tube length, large enough to allow free passage of sand, is r quired. In general, the harder or more brittle the scale, the smaller may be the opening.

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[2] The use of emulsifying oil as a cleaning agent for tubular equipment is covered by an application for U. S. letters patent.

(*c*) Sandblast the tubes. It has been found that ordinary labor, with reasonable care, can perform this operation satisfactorily without special equipment.

(*d*) Excess sand is blown from the tubes, the tubes inspected, and the unit closed and returned to service.

It should be emphasized that we have not found a measurable metal loss in a sandblasted tube. Sandblasted tubes have been removed from bundles, split longitudinally, and found to have a wall thickness over their entire length which was within normal manufacturing tolerances.

It is our opinion that the heat transfer in a sandblasted tube equals that for a new tube. In addition, qualitative, and scanty quantitative data indicate very definitely that, in many cases, the fouling rate is less for a sandblasted tube than for one cleaned by more conventional mechanical methods. This is believed to be due to the perfectly clean surface which is produced.

In our opinion, sandblasting as a supplementary procedure, should be considered when rodding, drilling, and other processes are the primary cleaning methods required. The reduction in fouling rate due to the complete removal of scale of course must be balanced against the additional cost, due to longer down time, and greater man-power requirements, as well as the necessity for additional cleaning equipment.

Fouling Rates and Cleaning Methods in Refinery Heat Exchangers

By R. C. BUTLER[1] and W. N. McCURDY, JR.,[1] LINDEN, N. J.

The three phases of heat-exchanger operation and maintenance which are covered in this paper are: rate of fouling (operating data from exchangers in various refinery services are used to correlate rate of fouling with time); reduction of fouling (the use of pressure and temperature control, blanketed feed tankage, feed stripping, and injection of additives as means to reduce fouling rates is discussed, as well as results obtained by refineries); cleaning methods (the mechanics and effectiveness of some methods of cleaning heat exchangers are discussed, the methods described being sandblasting, mechanical cleaning, and chemical cleaning).

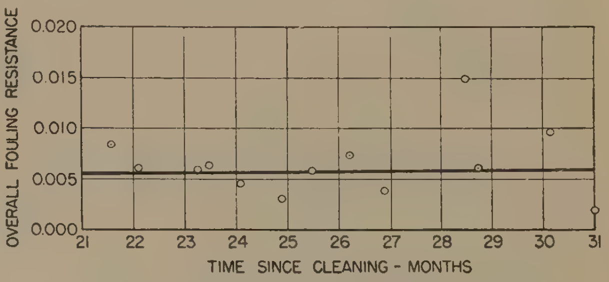

Fig. 1 Correlation of Fouling Resistance With Time, Pipestill Exchanger, Heavy Naphtha—Whole Crude
(Feed in tube side heated from 130 F to 180 F.)

Introduction

Fouling of exchangers and other refinery equipment costs refineries considerable sums each year in decreased capacity and lost heat, in addition to the actual cost of cleaning the equipment. Refinery experiences with rates of fouling, means of reducing fouling rates, and methods of cleaning exchanger equipment in various services are studied in determining the correct equipment design for new plants, and in solving fouling problems in existing plants.

Correlation of Fouling Resistance With Time

Since the economic optimum amount of heat-exchange surface, and the choice of providing block valves and by-passes to permit cleaning during plant operation are functions of the amount of deposits to be expected from the fluids undergoing heat exchange, it is essential that the designer have an idea of the rate at which the deposits will form under various conditions. To obtain this information, operating data from exchangers in various refinery services have been used as a basis for correlating fouling resistances with time.

Since it was not possible to determine from the data the exact amount of fouling occurring on each side (shell side or tube side) of the exchanger, a proportional amount of fouling was assigned to each side dependent on the appearance of the exchanger when it was taken down for cleaning. In most of the exchangers the fouling was almost entirely on one side only, and for the less dirty side, unless otherwise mentioned, the fouling has been assumed to be equivalent to a heat-flow resistance of 0.002 at the end of a run. Heat-flow resistance is the reciprocal of heat-transfer coefficient. The units are (hr)(deg F)(sq ft)/Btu.

The fouling curve shown in Fig. 1 is for a pipestill exchanger in which a heavy naphtha sidestream is being used to heat whole crude from 130 F to 180 F. It illustrates the small amount of fouling material which deposits out when the temperature is below the point where water in the crude will vaporize. The starting point on the graph is 21 months after cleaning. At the time the last data were taken the exchanger had not been cleaned for 36 months, and the coefficient of heat transfer was only a few per cent lower than immediately after cleaning.

Fig. 2 shows the fouling curves for a furnace oil reflux-whole crude exchanger on a crude pipestill over two 3-month periods. The fouling shown here, as in all of the figures, is over-all fouling, i.e., the sum of the shell-side and the tube-side fouling. Inspection after a run showed that the shell side (furnace oil reflux) was essentially clean, and most of the fouling had occurred on the tube (crude) side. The shell-side fouling resistance is assumed to be 0.002 after 6 months. Here the feed is being heated from 275 F to 330 F, and it fouls much more rapidly than it did at the lower temperature in Fig. 1. Although pressure is being held on the crude to prevent vaporization, some of the water in the crude flashes off at these temperatures, leaving a salt deposit in the exchanger tubes.

The two curves in Fig. 3, for a light gas oil to flashed crude exchanger, in which the flashed crude was heated from 315 to 355 F, show a decrease in fouling rate as compared to the whole-crude exchangers in Fig. 2. Flashed crude is whole crude which has been passed through a flash drum at reduced pressure (about 35 psi), where most of the water in the crude is flashed off. Following the flash drum, the salt in the flashed crude becomes practically dry, with the result that salt deposition at ascending temperature in hotter exchangers is minimized. It may be noted that after 3 months of operation, this exchanger showed an overall fouling resistance of 0.0085, as compared to a value of 0.0250 after 3 months for the whole-crude exchanger just ahead of the flash drum.

Curve A in Fig. 4 shows the rate of fouling for this same flashed-crude stream as it is being heated on to 450 F by reduced crude. The fouling rate in this exchanger is slightly higher than it was for the light gas-oil-flashed crude exchanger because the temperature of the flashed crude is higher. This exchanger was on stream for slightly more than 8 months before cleaning was necessary.

Curve B in Fig. 4 shows considerably less flashed-crude fouling than either Fig. 3, or curve A in Fig. 4, although the temperature (480 F outlet) is higher. This probably is due to decreased salt deposition, since the residual water present in the crude after leaving the flash drum is largely flashed out by the time the flashed crude is heated to this temperature. In the reduced-crude exchangers, the reduced crude (about 0.98 specific

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[1] Process Engineer, Standard Oil Development Company.

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society. Paper No. 48—A-117.

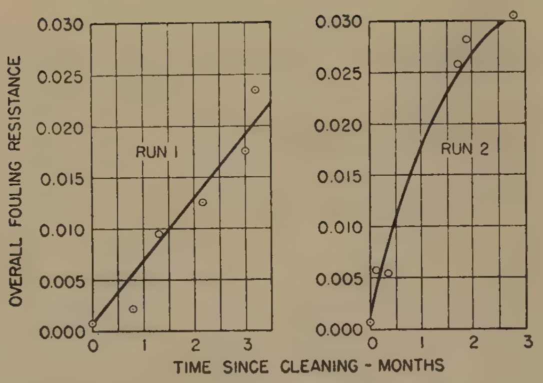

Fig. 2 Correlation of Fouling Resistance With Time, Pipe-still Exchanger, Furnace Oil—Whole Crude
(Feed in tube side heated from 275 F to 330 F.)

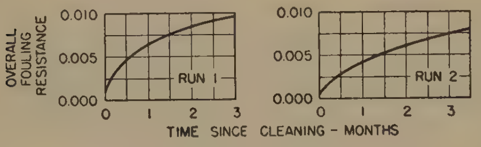

Fig. 3 Correlation of Fouling Resistance With Time, Pipe-still Exchanger, Light Gas Oil—Flashed Crude
(Flashed crude in tube side heated from 315 F to 355 F.)

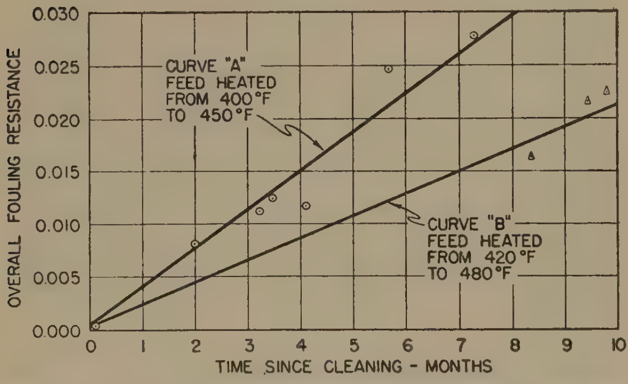

Fig. 4 Correlation of Fouling Resistance With Time, Pipe-still Exchanger, Reduced Crude—Flashed Crude
(Flashed crude in tube side being heated.)

gravity at 60 F), which is being cooled, although it is a heavy oil, fouls only slightly In general, it has been found that liquid heavy oils when below their coking point do not foul appreciably upon being cooled.

The three runs shown in Fig. 5 are representative of the fouling rate of a gas-oil feed which is being heated from 100 F to 200 F. These graphs indicate that this comparatively light gas-oil stream actually fouls more than heavier streams at higher temperatures.

Fouling in this case was caused primarily by the presence of dissolved oxygen in the feed stock which is pumped from open tankage. Where the feed tankage cannot be inert-gas-blanketed, stripping the feed stock to remove dissolved oxygen or the use of additives to control gum formation, as described in a later section, may be desirable.

The fouling curve for a slurry-feed exchanger on a catalytic-cracking plant is shown in Fig. 6. Here, as in Fig. 5, the fouling is believed to be due primarily to the presence of dissolved oxygen in the feed stock, which is heated from 185 F to 360 F in the shell side of the exchanger. The tube catalyst-in-oil (slurry) side of this exchanger does not foul appreciably in operation. The absence of fouling on the tube side is believed due to the scouring action of the slurry. It should be stated that in recent operations, as the slurry temperature has been increased to values of 675 to 700 F out of the desuperheating section of the tower, some difficulty with coke deposits on the tube sheets and in the tubes of the exchangers has been experienced.

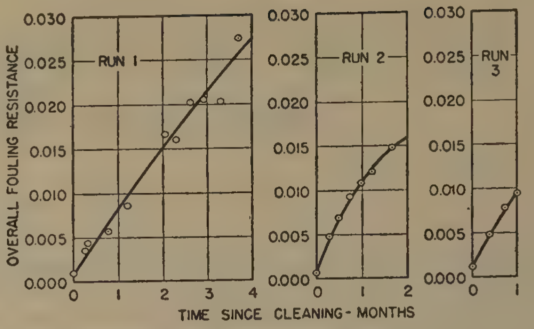

Fig. 5 Correlation of Fouling Resistance With Time, Catalytic-Cracking Unit Exchanger, Mid-Recirculating Reflux—Gas-Oil Feed
(Feed from open tankage heated in shell side from 100 F to 200 F.)

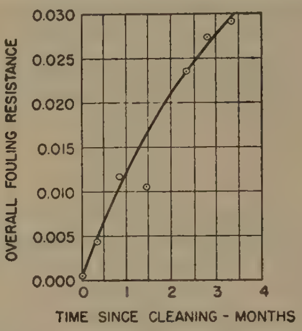

Fig. 6 Correlation of Fouling Resistance With Time, Catalytic-Cracking Unit Exchanger, Slurry—Gas-Oil Feed
(Feed heated in shell side from 185 F to 360 F.)

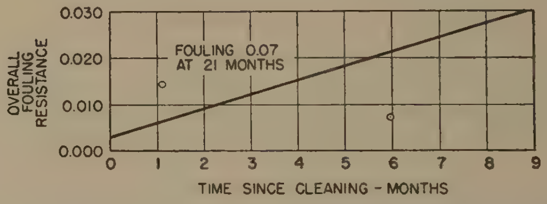

Fig. 7 Correlation of Fouling Resistance With Time, Gas-Absorption-Plant Exchanger, Lean Oil—Fat Oil

Fig. 7 shows the fouling in one of the lean oil-fat oil exchangers at a gas-absorption plant. This particular exchanger was on stream for slightly over 21 months between cleanings. By the end of that time the fouling resistance had become 0.070 and the over-all coefficient had dropped to about 40 per cent of the value immediately after cleaning.

Reduction of Fouling

In this portion of the paper, experiences by refineries with various fouling services, and the methods used to reduce the fouling are described. The methods used in one case may not apply directly to another case. Use of laboratory facilities to determine the nature of the deposits is desirable.

Fouling From Caustic Soda. The feed preheater of an isobutane tower was fouling. Analysis indicated caustic-soda deposits. Local water washing of the light hydrocarbon feed stream greatly decreased the rate of fouling.

Fouling From Salt in Crude. Salt deposited from crude, due primarily to vaporization of the water, has been a cause of decreased heat exchange and increased pressure drop in exchangers. Settling in feed tankage is desirable to reduce the quantity of salt solution, but is not economical beyond a certain point owing to the tankage required. Desalting by mechanical separation, such as brine addition to the crude to secure coalescence, followed by settling, or water addition followed by bed filters, is an acceptable process for some crudes. Good results may be secured by electrical desalters. Competition is resulting in decreased costs of the electrical method.

In view of the initial investments and operating costs involved in designs for new plants, studies must be made to determine if desalting equipment should be provided, or if pressure-temperature control of the crude is the economic choice. The latter method has been used in a number of refineries. Sufficient pressure is imposed on the crude to minimize vaporization of water and consequent salt deposition in the heat exchangers preceding a flash drum. When the crude has been preheated to a temperature of 300–350 F, the pressure is reduced immediately preceding the drum, and in the drum a large portion of the water is vaporized. Hot feed pumps are then used to pump the flashed crude through the subsequent exchangers and the furnace. By this procedure the fouling which otherwise would occur is greatly reduced. Calculations and refinery experience have been used in determining the optimum pressure-temperature relationship.

Fouling From Leached Brickwork. In an isobutylene-extraction plant, the sulphuric acid was being reconcentrated at about 300 F in a brick-lined concentrator. Subsequently, acid is mixed with hydrocarbons. Heat of reaction is removed by emulsion coolers. The coolers fouled so seriously that cleaning was necessary every 2 to 3 weeks. Analysis of the deposits indicated iron and aluminum salts resultant from leaching of the brickwork and from corrosion products. Emulsion velocities in the tubes were not a factor, as in test equipment fouling occurred at emulsion velocities of 16–18 fps. Reynolds numbers were about 2 cycles above the transition region. After equipment was installed to permit vacuum concentration of the acid at about 175 F, fouling of the exchangers greatly decreased and the interval between cleanings was extended to more than 3 months.

Fouling From Oxidation Products. Plant feed streams which previously have been processed, such as gas oils and naphthas, readily pick up oxygen in tankage. These streams deposit gums on the heat-exchanger surfaces, particularly when being heated through the region from 250 F to 450 F. In a hydroformer plant running a 200 F–270 F boiling range naphtha, it had been necessary to clean the feed exchangers every several weeks, as the pressure drop would increase to 100 psi during that time. To improve this situation, natural gas was used to blanket the feed tankage, and as a result it was feasible to extend the interval between cleanings to 60 days.

In some cases the feed to a unit consists of several mixed stocks, and blanketing of all feed tankage is not practical. Further, experience has indicated that blanketing with natural or inert gas is not totally effective, nor is the use of floating roof tankage. Stripping the unit feed, as it is charged to the unit, with an oxygen-free gas, including steam, is effective.

In determining if stripping is desirable,[2] the potential gum content and the oxygen content of the oil feed are important factors. The combination of these factors is an indication of the fouling tendency of the feed. The amount of deposit resulting from passing a sample of the feed through a glass coil submerged in a hot oil bath is a dependable means of determining if fouling of the plant heaters would be expected, and if feed stripping should be considered.

Relative to oxygen content, it was found that naphtha and gas oil in normal refinery handling could dissolve oxygen readily up to about 0.002 per cent by weight, and tests showed that this amount of oxygen would cause severe fouling. If the oxygen content is reduced to below 0.001 weight per cent in an oil, otherwise free from gums or sediment, little difficulty will occur from exchanger fouling. In one gas-oil-feed stripper, 24 ft of $^3/_4$-in. Raschig rings were used. Stripping with 10 to 20 cu ft of natural gas per barrel of feed reduced the oxygen content from 0.002–0.005 weight per cent down to less than 0.0004 weight per cent. Prior to stripping, it was necessary to reduce the unit feed rate at the end of a month due to pressure-drop limitations. After the feed stripper was put in service, the first year's operation resulted in negligible exchanger fouling.

Use of Bleed-Off in Recirculating Systems. In one type of gas absorption plant, the absorption oil consists mostly of recirculated oil, with small percentages of make-up oil as required. Lean (low-vapor-pressure) oil is contacted with wet gas, the heavier components of which are absorbed. The enriched oil is then passed through heat exchangers to stills where the absorbed gases are removed and recovered. The hot stripped oil is returned through heat exchangers to the absorption tower. Considerable fouling difficulties are often experienced in this type of recirculating absorption plant. Extensive laboratory and plant studies have been made. The fouling compounds form as a result of thermal oxidation of the unsaturates in the incoming-gas stream. The oxidation is stepwise. In the intermediate stages of the oxidation, the partially oxidized materials are soluble in the absorption oil and have similar distillation characteristics, which makes it difficult to separate them by distillation. These partially oxidized materials are readily soluble in concentrated sulphuric acid but not in caustic. As oxidation progresses, materials are formed (probably salts of organic acids or resin) which precipitate and adhere to metallic surfaces, particularly to the heating and cooling surfaces of the exchanger equipment.

Studies indicated that fouling was materially reduced by the following:

1 Eliminating oxygen or unsaturates from the system.

2 Contacting the recirculated absorption oil with concentrated sulphuric acid to remove the partially oxidized materials.

3 Maintaining a high make-up rate and discard of absorption oil to remove the partially oxidized materials before the oxidation has progressed to a point where precipitation takes place.

The refinery found that it was not economically feasible to remove oxygen or unsaturates owing to the large amount of equipment and chemicals required (about 25 lb of sulphuric acid per bbl of oil). Contacting the absorption oil with strong sulphuric acid to remove the partially oxidized materials was effective but expensive. For this particular refinery, it was feasible to use the third method of a high make-up and discard rate, since the discarded oil could be used efficiently as reflux in an adjacent pipestill. The discard rate is 700 bbl per day. It should be mentioned that the use of inhibitors was not successful.

Use of Additives. No definite information is available on the efficacy of adding gum inhibitors. It is reported that one company, by the injection of an inhibitor, secured reduced fouling of the feed side of the slurry-feed exchangers of a catalytic-cracking plant.

Addition of Chlorine to Cooling Water. Coolers and condensers foul on the water side causing increased pressure drops, inability to pass sufficient water through the exchangers, and decreased heat-transfer coefficients. Frequently fouling is due to algae growth. The algae growth adheres to the tube walls and forms a structure for the accumulation of silt. Control of the marine growth also is desirable relative to the performance of cooling towers. Various chemicals, including discarded sulphuric acid, copper sulphate, sodium hypochlorite, and chlorine have been employed to control bacteria activity and marine growth. Of these chemicals, chlorine is considerably preferable. Heat-transfer coefficients before and after chlorination for a shell-and-tube-type overhead naphtha condenser at a catalytic-cracking plant are indicated in Table 1.

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[2] "Non-Fouling Refinery Charge Stocks," by L. F. Schimansky, *Petroleum Refiner*, vol. 25, no. 12, 1946, pp. 119–121.

TABLE 1 EFFECT OF CHLORINATION

Before chlorination		After chlorination	
Days since cleaning	Btu/-(hr sq ft deg F)	Days since cleaning	Btu/-(hr sq ft deg F)
0	42	0	43
8	30	16	36
53	25	36	41
100	23	64	42
153	22	112	41

From Table 1 it can be seen that there was practically no decrease in the heat-transfer coefficient during 4 months when the water was chlorinated. The water velocities in the tubes normally are 3 to 5 fps. Experience before and after chlorination in various refineries has shown that, in general, the benefits to be derived from chlorination justify the investments and operating costs necessary. New refineries usually include chlorination facilities, and in many existing refineries means of chlorination have been installed. In recirculated water systems, chlorine is added continuously to result in a residual chlorine content (in the water returning to the cooling tower) of 0.5 to 1 ppm. In once-through systems (such as salt-water systems) satisfactory control and economy of chlorine are obtained by intermittent injection. During the injection period of about 1 hr every 8 hr, sufficient chlorine is injected to result in about 1.5 ppm residual chlorine.

pH Control of Cooling Water. In recirculated (cooling-tower) water systems, the formation of carbonate scales may occur. Salt deposits of this type may be controlled by the addition of various chemicals. One refinery employs caustic and Hagan phosphates. Another has found that sulphuric acid is better for its type of water. In general, control of pH to a value of about 8 is desirable to control salt-scale formation.

Cleaning Methods

Tube bundles are cleaned in place, and by withdrawing them from the shell. When the outsides of the tubes are to be cleaned, in-place cleaning requires the circulation of a solvent. Solvent circulation has limitations, since the correct fluid must be circulated to secure effective cleaning. Usually, a local or portable tank and pump and connecting lines are required. The major advantage frequently is that this method reduces down time of the plant. An extra 6 to 12 hr down time of a large plant will mean considerable monetary loss. Another advantage is that usually cleaning in place is less expensive with regard to labor and material costs. The disadvantage of solvent in-place cleaning is the uncertainty of results. When the inner surfaces of the tubes are to be cleaned, in-place cleaning may be done using either mechanical or chemical means.

When tube bundles are cleaned by withdrawing them from the shell, the plant down time may be kept to a minimum by having a spare tube bundle ready to install in the place of the dirty one. The fouled bundle then may be cleaned locally, or transported to a cleaning shop. Many methods are used for out-of-shell cleaning. These methods include mechanical cleaning with cleaning bars, scrapers, rods, drills, and brushes; also included are cleaning by immersion in a tank of solvent, sandblasting, the use of hydraulic jets or steam jets, and controlled burning when the deposit is mostly carbon. The choice of in-place or out-of-shell cleaning depends upon the nature of the deposit, the degree of success with the method when previously used, and the local preference of the cleaning-gang foreman and supervising personnel.

One refinery has classified deposits into seven types, and employs a recommended cleaning method for each type, as indicated in Table 2.

TABLE 2 TYPES OF DEPOSITS AND CLEANING PROCEDURES

Type of deposit	Recommended cleaning procedure
Diolefin polymer on steel-tube bundle	Contact with 90–98 per cent sulphuric acid in vat for 2 hr at 175 F–200 F; follow by cold-water wash
Residuum or tar	Soak in circulating hydrocarbon solvent at elevated temperatures
High carbon low oil content	Soak in hot gas oil (6–8 hr), followed by 8 hr in 0.5 per cent water solution of Oakite Railroad Cleanser; final removal by 175-psi water jet
Low carbon high oil content on steel-tube bundle	Soak in 98 per cent sulphuric acid at 150 F, followed by water wash; finally soak in 0.5 per cent water solution of Oakite railroad cleanser
Iron, sulphur, carbon deposits	Inhibited HCl followed by soaking in 0.5 per cent Oakite Railroad Cleanser at 210 F for 12–18 hr
Butyl rubber	Varsol (hydrocarbon solvent) at atmospheric pressure
Water scales	Inhibited HCl at 100 F to 150 F for 2–6 hr; periodic addition of acid if necessary

A study of this refinery's recommended methods indicates that the principles may be used for either in-place or out-of-shell cleaning, with the exception of the use of water jets.

For out-of-shell cleaning of nonferrous bundles, one refinery prefers acid-pickling for general use. The acid bath consists of 3000 gal of hot salt water and 800 gal of 35 Bé sulphuric acid, i.e., a 10 per cent concentration of acid. Tube bundles are immersed in this bath for a period of 16 hr during which time the bath is kept at 150 F by steam heat. Bundles are then washed with water to remove acid. On one test the interior of the tube bundles after immersion and wash showed 75 per cent bright metal exposed. It costs approximately $150 to clean a 400-sq ft bundle by this method, exclusive of removal cost.

Metal loss based upon weighing two pieces of Admiralty metal tubing, one new and one used indicated a metal loss of 0.0042 grams per sq in. when immersed in the same acid as the bath for 24 hr at 160 F.

Some refineries have found crude sodium sulfonates effective as cleaning agents. With sulphonate detergents, in-place cleaning may be used. The emulsifying agent is effective in cleaning exchangers and other equipment which are fouled with oil, oily residues, or deposits composed of carbon or mineral matter bound by oil or oily residues. In general, the crude sulphonates are mixed in varying concentrations with water containing 2 or 3 per cent sodium carbonate and circulated through the equipment. (The crude sulphonates are produced as a by-product from residual acid oil remaining after heavy acid treatment of a coastal distillate phenol extract. The residual acid oil is neutralized with sodium hydroxide at 210 F, and the crude sulphonate, composed of approximately equal parts of oil, water, and sulphonates, is separated from the bulk of the oil by the addition of 5–10 per cent of water.) The refineries feel that considerable savings are effected by the use of this method because it eliminates removing the tube bundle and the resultant down time.

In-place cleaning by the solution method has been tried but has not proved satisfactory in all cases. Before in-place cleaning is attempted, it is necessary to have a thorough knowledge of the exact deposit to be removed. A comparison of solution cleaning and mechanical cleaning would favor the solution method, assuming the solution employed was effective.

Scale deposits from cooling water are removed readily by in-place solution-cleaning methods. Either an outside company or the refinery facilities may be employed. The method is to circulate inhibited HCl at about 150 F through the tubes of the condenser or cooler. This is continued for from 2 to 6 hr, or until CO_2 evolution is small. After the acid treatment, water wash is employed.

During comedown of a plant, it is sometimes possible to accomplish partial cleaning by circulating a hot fluid having solvent action through the exchangers before they have cooled off and the deposits become dry.

Means of out-of-shell cleaning which employ hydraulic jets and controlled oxidation of the carbon deposits have not been investigated completely. One refinery reports that 300 to 400-psi water jets do an excellent job of outside tube cleaning. The exchanger bundle is placed on a car and moved slowly on tracks past the high-pressure water jets. Another refinery has reported successful cleaning of a tube bundle covered with hard thick coke. The bundle was moved into an oven, and the carbon burned off successfully. It is indicated that careful control of oxygen and burning rate will be necessary when the burning method is resorted to.

Sandblasting is a cleaning method which is finding increasing favor. It may be used to clean the inside of the tubes of an inplace bundle, or either side of the bundle when it is withdrawn from the shell. It is faster and more effective than the old method of washing with hose streams, punching, and drilling. A good sandblasting job on a badly fouled bundle will clean the unit so effectively that it will perform like new.

A sandblast team normally consists of two men, i.e., a nozzleman and a machine attendant. Two or more teams can work effectively on one exchanger bundle without interference. A team can be taught the fundamentals within a few hours, and within a few days the average man will be well qualified. No special skills are prerequisite for the training of sand blasters and these men are obtained from the labor gangs.

The nozzle is usually held from 2 to 14 in. from the object to be blasted. Experience indicates the proper "shooting distance." A sweeping rotary motion is generally employed until bright metal appears. In cleaning the internal surface of exchanger tubes, the nozzle is placed at the end of the tubes so that the bore of the nozzle is concentric with the inside diameter of the tube. Experience has shown that it is seldom necessary to blast from both ends of a bundle to remove all foreign matter. It is important that coke and carbonate-type scales be dry to prevent plugging. When a tube is plugged, as sometimes happens, the sand can usually be removed by air pressure, punching with a rod, or, in rare cases, drilling. Weather conditions have little effect on operations, but it is mandatory that the abrasive be thoroughly dry and the object to be blasted reasonably dry.

An average of one hundred $\frac{5}{8}$-in-OD × 16-BWG × 144-in. tubes (on square pitch) can be cleaned by a two-man team in 1 hr. The length of a tube does not greatly influence cleaning time in so far as internal surface is concerned.

The abrasive recommended for optimum results in cleaning exchanger tubes, internally and externally, is a sharp, angular sand equal to washed Colorado River (Texas) sand which is screened through a $\frac{1}{4}$-in. mesh. All sand must be thoroughly dry to reduce the inherent tendency of the hose to plug. A gas-fired rotary-kiln drier may be used, and the dry sand stored in hoppers for distribution. The practice of using reclaimed sand is not generally permitted owing to the presence of carbonates, oxides, sulphides, oil, and coke. Compressed air at about 100 psi is used at the rate of 200 cfm per team.

ACKNOWLEDGMENT

The authors wish to express their appreciation of the cooperation of Mr. M. J. Lang and Mr. G. K. Simpson of the Esso Standard Oil Company, Bayway Refinery, who supplied the test data relative to fouling rates used in the paper.

TABLE 3 PHYSICAL AND PROCESS DATA FOR EXCHANGERS FOR WHICH FOULING RATES ARE REPORTED

Fouling data shown in	Fig. 1	Fig. 2	Fig. 3	Fig. 4	Fig. 5	Fig. 6	Fig. 7
Service of unit	Pipestill, hvy naph-whole crude	Pipestill, furnace oil-whole crude	Pipestill, light gas oil-flashed crude	Pipestill, red. crude-flashed crude	Catalytic cracker, circulating reflux-gas oil	Catalytic cracker, slurry-gas oil	Gas absorption plant, lean oil-fat oil
Surface (external), sq ft	1100	4400 4 shells in series	2400 2 shells in series	1535 ($\frac{1}{2}$ of total) internal surface)	1060	1200	1360
Tube size, OD (in.), BWG, length (ft)	$1\frac{1}{2}$-10-16	$1\frac{1}{2}$-10-16	$1\frac{1}{2}$-10-16	$2\frac{1}{2}$-8-30	$1\frac{1}{4}$-12-16	$1\frac{1}{2}$-10-16	$\frac{5}{8}$-16-12-1[a]
Tubes per shell	182	182	199		208	199	660
Tube passes	4	4	6		4	6	4
Tube pitch and layout	$1\frac{7}{8}$-square	$1\frac{7}{8}$-square	$1\frac{7}{8}$-square	(See Note)	$1\frac{9}{16}$-square	$1\frac{7}{8}$-square	$\frac{7}{8}$-square
Shell ID, in.	33	33	35		29	35	34
Baffle pitch, in.	$13\frac{1}{8}$	$15\frac{7}{8}$	$5\frac{15}{16}$		$6\frac{1}{2}$	6	$5\frac{1}{2}$
Baffle type	Segmental	Segmental	Segmental		Segmental	Segmental	Segmental

	Fig. 1		Fig. 2		Fig. 3		Fig. 4		Fig. 5		Fig. 6		Fig. 7	
	Shell side	Tube side	Shell side	Tube side	Shell side	Tube side	Tube side	Tube side	Shell side	Tube side	Shell side	Tube side	Shell side	Tube side
Fluid circulated	Heavy naphtha	Whole crude	Furnace oil	Whole crude	Light gas oil	Flashed crude	Reduced crude	Flushed crude	Gas oil	Mid-recirculation reflux	Gas oil	Slurry	Lean oil	Fat oil
Quantity fluid circulated, lb/hr	318000	532000	313000	558000	167000	564000	70000	260000	252000	206000	121000	170000	87000	159000
Approx temp (in and out), deg F	279–229	134–167	470–405	268–314	364–323	316–330	570–450	426–457	87–223	461–344	217–389	590–484	209–176	145–162
Approx density, lb/cf	49	52	52	52	53	52	59	55	55	55	55	59	51	46
Mass velocity, lb/sec ft^2	108	390	90	410	124	570	250	310	197	190	85	170	160	200

[a] Dimension given, 12 ft 1 in.

NOTE: This is a Tubeflo type of exchanger. The colder fluid flows through 9 tubes in parallel and the hotter fluid through 3 in parallel.

Rates of Fouling and Cleaning of Unfired Heat-Exchanger Equipment

By J. H. WEILAND, JR.,[1] R. C. McCAY,[2] and J. E. BARNES[1]

This paper presents heat-exchanger performance data and describes exchanger cleaning procedures during the period 1943–1947 at the Port Arthur Works of The Texas Company. The performance data are presented in the form of tables and graphs which show the rates of over-all heat-transfer and exchanger fouling, and the changes in these rates after bundle cleansing.

In an effort to provide useful data on fouling and methods of cleaning of unfired heat-exchange equipment, data were developed from test runs and survey periods. The types of heat-exchange services which were considered are as follows: (1) Cycle fuel-oil-heated naphtha reboilers; (2) depropanizer reflux condensers; (3) absorption oil coolers; and (4) gas oil-catalyst slurry exchangers.

The operating data and the calculated values of the heat duties, over-all heat-transfer coefficients, film resistances, and total fouling resistances are presented in Tables 1 to 4, inclusive. Graphical presentations of the over-all heat-transfer coefficients and fouling resistances are shown in Figs. 1 to 4, inclusive. In all cases, only data with comparable operating conditions were used. The flow rates and entering temperatures, as well as unit conditions, were maintained as constant as possible throughout the report data. All calculations were made on the basis of correlations available to the authors' company.

Types of Fouling

A brief discussion of the types of fouling in each of the services mentioned is given as follows:

Cycle Fuel-Oil-Heated Naphtha Reboilers. Inside the tubes, coke is deposited in loose form, and usually may be removed by punching, blowing, sandblasting, and washing. The shell-side deposit is usually not heavy and does not result in plugging between tubes. Water which may enter the shell side of the reboilers has caused rust formation.

Depropanizer Reflux Condensers. The insides of the tubes remain fairly clean as long as sufficient clean water is available. During prolonged periods of dry weather the sediment content of the cooling water increases, resulting in additional tube-side fouling. Light hydrocarbons leaking into the cooling-tower return streams give rise to a stable emulsion which tends to plug the tubes. This is usually removed by washing each tube individually with a stream of high-pressure water.

The shell-side deposit consists of a loose flaky scale, composed mainly of metal sulphides. Although fairly easily removed from the outside of the bundles by washing, the scale plugs the spaces between the tubes and must be removed by hand-scraping methods.

Absorption Oil Coolers. The sediment deposits in the tubes are usually removed by washing each tube separately with a stream of high-pressure (150-psi) water. The shell-side fouling material is a loose scale which is removed to a large extent by washing, and the remainder is removed by scraping. The shell sides of the bundles are washed with water one time between semiannual test and inspection down periods. This prevents the scale from plugging between the tubes.

Gas Oil-Catalyst Slurry Exchangers. The fouling material on the shell side of the bundles consists of a viscous gummy sludge which is held tenaciously to the surfaces of the tubes. This sludge is acidic in nature, but if washed with caustic solution, the acidic components are neutralized, and the sludge disperses into hard particles which may be removed by caustic circulation and subsequent water washing. If the sludge is allowed to remain on the tubes over an extended period it tends to harden and seemingly is more difficult to remove by caustic washing. The practice at the Port Arthur Works is to caustic-wash at monthly intervals.

Cleaning Procedure

With but one exception, the exchanger-bundle cleaning methods were and still are mechanical. The general cleaning procedure is outlined briefly as follows:

After the unit has been shut down or the exchange equipment by-passed, and before the exchanger shell is unheaded, the shell side of the bundle is washed thoroughly with water and steamed. This serves to wash away most of the loose and soluble deposits, and to soften the remaining deposits. When both sides of the exchanger have been drained and the shell side washed, the exchanger heads are removed, and the bundle is withdrawn from the shell.

If the tube side is partially fouled or plugged, an exchanger-tube scraper is used to ream the tubes. This scraper consists of a hollow shaft with an outer diameter slightly less than the inner diameter of the exchanger tubes. A steel cutting bit is welded to one end of the shaft, and just below this bit are wire springs which scrape the walls of the tube. Either liquid (oil, or water), or gas (steam or air) may be charged through the shaft as the scavenger agent. The flushing medium is discharged through the shaft at the points where the springs are attached. The shaft rotates at about 700 rpm while it is being pushed through the tubes. In the event that a tube is plugged to such an extent that the tube scraper cannot be forced through, special vibrating equipment is attached to the shaft. The shaft rotation is increased to 3500 rpm while, at the same time, light hammer blows are applied to the shaft at the rate of 1900 times a minute.

For cleaning the shell side of the bundles, hand scrapers are utilized. These scrapers are thin bars of steel, 1 in. wide, $^1/_8$ in. thick, and long enough to extend through the bundle. They usually are serrated on one side. The scrapers are inserted between the rows of tubes and are worked back and forth the length of the bundle. The deposits are scraped off and fall onto baffles or other tubes. The bundle is blown with air or steam periodically during the scraping operation in order to remove the loosened deposit. The scraping and blowing are continued until all scale and deposit have been removed from the shell side of the tubes.

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[1] The Texas Company, Port Arthur Works, Port Arthur, Texas.

[2] The Texas Company, New York, N. Y.

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society. Paper No. 48—A-112.

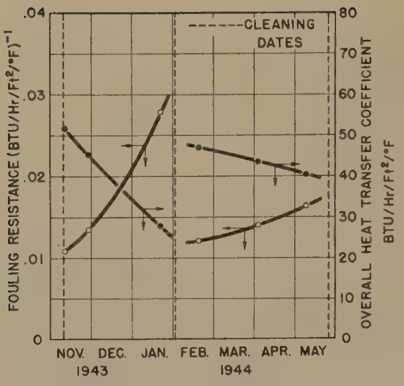


FIG. 1 CYCLE FUEL-OIL-HEATED NAPHTHA REBOILERS; FOULING RESISTANCE AND OVER-ALL HEAT-TRANSFER COEFFICIENT VERSUS TIME

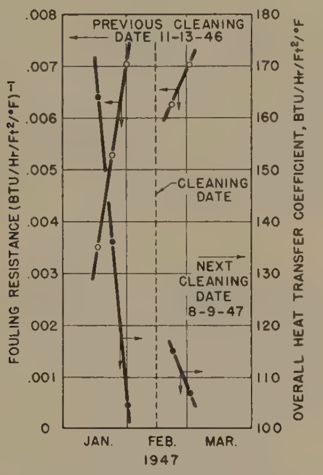


FIG. 2 DEPROPANIZER REFLUX CONDENSERS; FOULING RESISTANCE AND OVER-ALL HEAT-TRANSFER COEFFICIENT VERSUS TIME

The one example of chemical cleaning which is covered in thi paper is the use of sodium-hydroxide solution (5–10 deg Baumé in cleaning the shell side (gas oil) of gas oil-catalyst slurry exchangers. The four exchangers are cleaned, two at a time, without shutting down the unit or pulling the bundles. The causti is circulated through the shell, while steam is passed through th tubes. After the caustic circulation, the bundles are washed an steamed water-dry in about $1^1/_2$ hr. It was found that if th caustic temperature were kept near its boiling point bette cleaning resulted. The washing period was originally 24 hr long but was gradually reduced to 6 hr.

An inspection of the graphs of the rates of fouling and change in heat-transfer coefficient reveal the expected general trends i.e., increase in fouling and decrease in transfer coefficient wit time.

CONCLUSION

In conclusion, it should be stated that the mechanical method of cleaning which have been used appear to be adequate, considering the structural characteristics of bundles. The man-hour requirements for cleaning the outside of the tube bundles by scrap

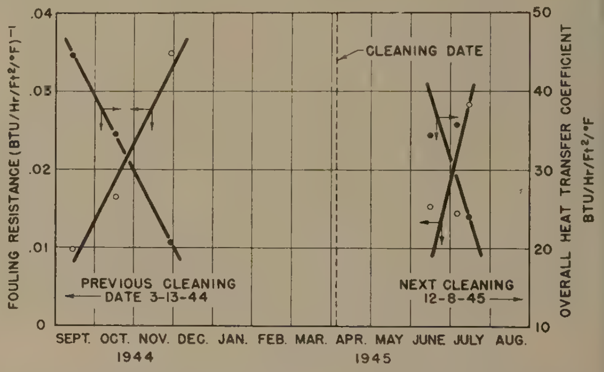


FIG. 3 ABSORPTION OIL COOLERS; FOULING RESISTANCE AN OVER-ALL HEAT-TRANSFER COEFFICIENT VERSUS TIME

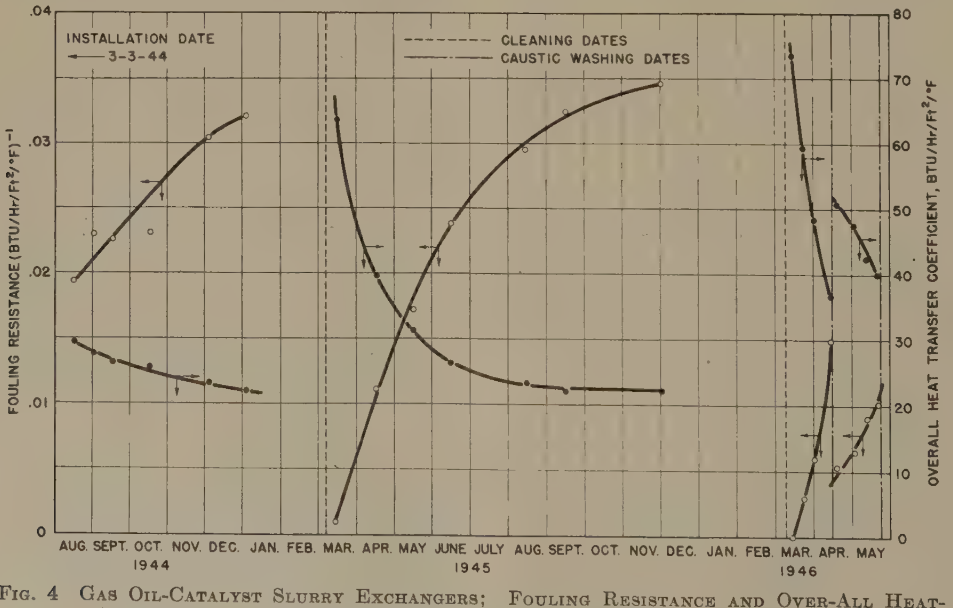
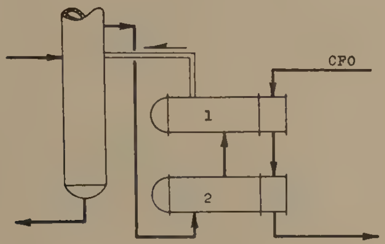


FIG. 4 GAS OIL-CATALYST SLURRY EXCHANGERS; FOULING RESISTANCE AND OVER-ALL HEAT-TRANSFER COEFFICIENT VERSUS TIME

TABLE 1 CYCLE FUEL-OIL-HEATED NAPHTHA REBOILERS (2)

Date	Flow Materials		Flow Rates - Barrels/Hr.		Terminal Temperature of Fluid Stream, °F		Corrected LMTD, °F	Overall Heat Trfr. Coeff, BTU/Hr/Ft²/°F	Film Resistances (BTU/Hr/Ft²/°F)$^{-1}$		Overall** Fouling Resistance (BTU/Hr/Ft²/°F)$^{-1}$	Heat Duty BTU/Hr
	Shell-side	Tube-side	Shell-side	Tube-side	Shell-side	Tube-side			Shell-side	Tube-side		
11-9-43					Exchangers Cleaned							
11-10-43	Crkd. Naph.	CFO	585	385	408 → 451	424 ← 637	62.8	51.6	.00530	.00327	.01083	17,815,000
11-28-43	Crkd. Naph.	CFO	620	386	392 → 440	415 ← 640	75.6	45.2	.00520	.00324	.01368	18,863,000
1-21-44	Crkd. Naph.	CFO	632	535	412 → 448	482 ← 639	119.0	27.8	.00520	.00290	.02790	18,206,000
2-2-44					Exchangers Cleaned							
2-19-44	Crkd. Naph.	CFO	544	324	415 → 448	426 ← 647	60.0	47.0	.00540	.00370	.01220	15,518,000
4-3-44	Crkd. Naph.	CFO	578	346	418 → 449	434 ← 650	67.7	43.6	.00530	.00354	.01406	16,284,000
5-8-44	Crkd. Naph.	CFO	573	432	419 → 453	445 ← 625	74.1	40.4	.00530	.00301	.01644	16,499,000
5-25-44					Exchangers Cleaned							



	1	2
Tube Passes	10	10
Shell Passes	1	1
Tubes: No	852	900
Size	1"x12'	1"x12'
Gauge	14	14
Pitch	1¼"□	1¼"□
Outside Area	2676 Ft^2	2827 Ft^2
Baffle Space	2'10"	2'10"
Installation Dates	6-2-41 2-18-44	12-5-38
Material	Steel	Steel

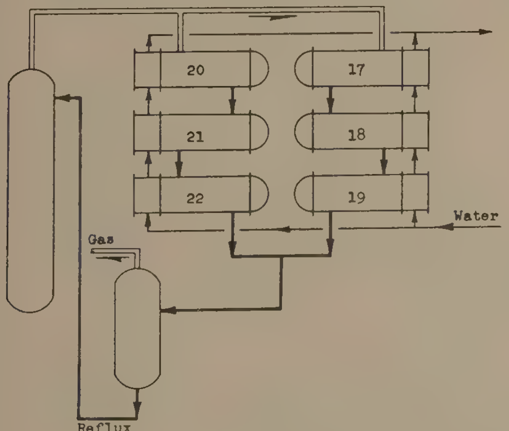
Flow Material Properties	CFO	Naphtha
Avg.Gravity, °API	5.0	58.0
Avg.Viscosity, Centipoises*	.1	--

*At Average Film Temperature of 525°F

**Includes Tube Thermal Resistance of .00028 (BTU/Hr/Ft²/°F)$^{-1}$

TABLE 2 DEPROPANIZER REFLUX CONDENSERS (6)

Date	Flow Materials		Flow Rates Shell Thousand of Cu Ft/Hr Gas	Flow Rates Barrels /Hr Reflux	Flow Rates Tube-GPM	Terminal Temperatures of Fluid Streams, °F		Corrected LMTD, °F	Overall Heat Trfr. Coeff, BTU/Hr/Ft²/°F	Film Resistances (BTU/Hr/Ft²/°F)$^{-1}$		Overall** Fouling Resistance (BTU/Hr/Ft²/°F)$^{-1}$	Heat Duty BTU/Hr
	Shell-side	Tube-side				Shell-side	Tube-side			Shell-side	Tube-side		
Prior Cleaning Dates:	11-13-46: Nos. 17, 18, 19; 8-30-46: No. 20; 12-15-46: No. 21; 8-19-46: No. 22												
1-17-47	C_3-C_2	Water	381	569	700	110 → 98	107 ← 74	5.96	164	.0012	.00139	.00351	12,275,000
1-24-47	C_3-C_2	Water	383	580	1040	111.5 → 99.5	108 ← 84	7.33	136	.0011	.00097	.00528	12,489,000
1-31-47	C_3-C_2	Water	305	580	662	114 → 99.5	111 ← 73	9.60	104.4	.0011	.00145	.00703	12,588,000
2-14-47						Exchangers Cleaned							
2-22-47	C_3-C_2	Water	428	571	686	115 ← 101	112.5 ← 76.5	8.60	115	.00105	.00139	.00626	12,360,000
3-1-47	C_3-C_2	Water	451	581	734	115 → 98.5	111 ← 76	9.63	106.7	.00105	.00133	.00701	12,855,000
8-9-47						Exchangers Cleaned							



	All
Tube Passes	4
Shell Passes	1
Tubes: No	884
Size	3/4x12
Gauge	16
Pitch	15/16"□
Outside Area	2083
Baffle Space	6-7/16"

Flow Material Properties

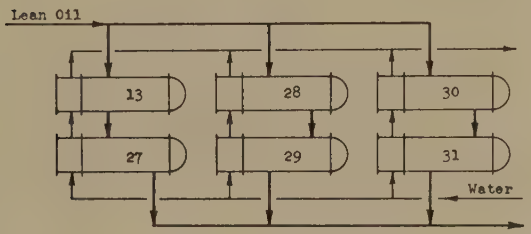
Avg.Lbs/MCFH OH Gas = 99.5
Avg.Lbs/Bbl. Reflux = 172

	17	18	19	20	21	22
Installation Date	7-24-45	3-14-46	10-30-45	12-15-46	8-30-46	10-30-45
Material	Admiralty Lined Steel	Cu-Ni	Cu-Ni	Cu-Ni	Cu-Ni	Admiralty

** Includes Tube Thermal Resistance of .00039 (BTU/Hr/Ft²/°F)$^{-1}$

TABLE 3 ABSORPTION OIL COOLERS (6)

Date	Flow Materials Shell-side	Flow Materials Tube-side	Flow Rates Shell- Barrels /Hr	Flow Rates Tube- GPM	Terminal Temperatures of Fluid Streams °F Shell-side	Terminal Temperatures of Fluid Streams °F Tube-side	Corrected LMTD, °F	Overall Heat Trfr. Coeff. BTU/ Hr/Ft²/°F	Film Resistances Shell-side (BTU/Hr/Ft²/°F)⁻¹	Film Resistances Tube-side (BTU/Hr/Ft²/°F)⁻¹	Overall** Fouling Resistance (BTU/Hr/Ft²/°F)⁻¹	Heat Duty BTU/Hr
Prior Cleaning Dates:	3-13-44: Nos. 13, 27; 2-25-44: Nos. 30, 31; 5-19-43: Nos. 26, 29											
9-13-44	Lean Oil	Water	1254	668	106 → 86	88.9 ← 78.6	10.8	44.5	.011	.00206	.00981	3,440,000
10-16-44	Lean Oil	Water	1402	586	106.5 → 88	88 ← 75.9	14.4	34.5	.0106	.00242	.01646	3,550,000
11-28-44	Lean Oil	Water	1215	481	106.0 → 86	79.8 ← 66	22.6	20.6	.0115	.00314	.03470	3,320,000
4-4-45 (3-14-45 for Nos. 30, 31)					Exchangers Cleaned							
6-15-45	Lean Oil	Water	1260	496	103.0 → 92.9	93.3 ← 85.8	7.55	34.4	.0115	.00272	.01530	1,860,000
7-5-45	Lean Oil	Water	1253	499	105.0 → 93.5	94.9 ← 87.0	7.70	35.7	.0112	.00272	.01445	1,970,000
7-14-45	Lean Oil	Water	1260	524	105.0 → 93.0	90.4 ← 82.5	12.0	24.0	.0113	.00262	.02834	2,063,000
8 (30, 31) 12-26-45 (28, 29) 31 (13, 27)					Exchangers Cleaned							



	13	27-31
Tube Passes	4	6
Shell Passes	2	1
Tubes: No	402	528
Size	3/4x12	3/4x12
Gauge	16	16
Pitch	1"□	1"□
Outside Area	947	1244
Baffle Space	8"	6"

Absorption Oil Properties	
Avg. Gravity, °API	42
Avg. Viscosity, Centipoises*	1.5

*At Average Film Temperature of 95°F.

		27	28	29	30	31
Installation Date	5-1-41	6-1-41	4-18-41	4-18-41	4-18-41	4-18-41
Material			Red Brass			

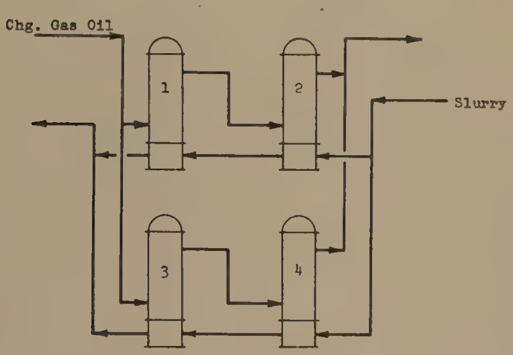
**Includes Tube Thermal Resistance of .00009 (BTU/Hr/Ft²/°F)⁻¹

TABLE 4(*a*) GAS OIL-CATALYST SLURRY EXCHANGERS (4); PRECAUSTIC WASH

Date	Flow Materials Shell-side	Flow Materials Tube-side	Flow Rates Barrels/Hr Shell-side	Flow Rates Barrels/Hr Tube-side	Terminal Temperatures of Fluid Streams °F Shell-side	Terminal Temperatures of Fluid Streams °F Tube-side	Corrected LMTD, °F	Overall Heat Trfr. Coeff. BTU/ Hr/Ft²/°F	Film Resistances Shell-side (BTU/Hr/Ft²/°F)⁻¹	Film Resistances Tube-side (BTU/Hr/Ft²/°F)⁻¹	Overall** Fouling Resistance (BTU/Hr/Ft²/°F)⁻¹	Heat Duty BTU/Hr
3-3-44					Installation							
8-15-44	Chg. Gas Oil	Slurry	681	2840	105 → 335	463 ← 509	250	29.3	.00631	.00684	.01926	26,200,000
9-1-44	Chg. Gas Oil	Slurry	740	2840	105 → 320	461 ← 507	259	27.5	.00655	.00684	.02301	25,400,000
9-16-44	Chg. Gas Oil	Slurry	485	2710	100 → 340	457 ← 505	246	26.1	.00870	.00705	.02255	22,885,000
10-16-44	Chg. Gas Oil	Slurry	615	2000	97 → 325	450 ← 508	256	25.4	.00770	.00865	.02305	23,300,000
12-3-44	Chg. Gas Oil	Slurry	819	2550	95 → 285	491 ← 537	311	23.0	.00640	.00664	.03046	25,580,000
1-3-45	Chg. Gas Oil	Slurry	861	2500	90 → 273	481 ← 528	318	21.8	.00662	.00724	.03204	24,700,000
3-6-45					Exchangers Cleaned							
3-13-45	Chg. Gas Oil	Slurry	810	2090	95 → 430	443 ← 548	204	63.6	.00514	.00965	.00092	46,200,000
4-15-45	Chg. Gas Oil	Slurry	864	2160	90 → 364	475 ← 560	275	39.7	.00553	.00865	.01102	38,900,000
5-15-45	Chg. Gas Oil	Slurry	850	2060	95 → 334	484 ← 561	298	31.3	.00593	.00885	.01722	33,300,000
6-15-45	Chg. Gas Oil	Slurry	802	2200	100 → 320	496 ← 558	308	26.2	.00616	.00825	.02379	28,800,000
8-14-45	Chg. Gas Oil	Slurry	842	2340	105 → 299	500 ← 554	318	23.3	.00615	.00725	.02950	26,500,000
9-15-45	Chg. Gas Oil	Slurry	848	2200	95 → 287	500 ← 556	330	22.0	.00622	.00684	.03244	26,000,000
12-1-45	Chg. Gas Oil	Slurry	1002	2450	90 → 257	487 ← 538	334	22.0	.00569	.00644	.03451	26,250,000
2-19-46					Exchangers Cleaned							

TABLE 4(*b*) GAS OIL-CATALYST SLURRY EXCHANGERS (4); CAUSTIC WASH

Date	Flow Materials Shell-side	 Tube-side	Flow Rates - Barrels/Hr Shell-side	 Tube-side	Terminal Temperatures of Fluid Streams °F Shell-side	 Tube-side	Corrected LMTD, °F	Overall Heat Trfr. Coeff. BTU/Hr/Ft²/°F	Film Resistances Shell-side $(BTU/Hr/Ft^2/°F)^{-1}$	 Tube-side $(BTU/Hr/Ft^2/°F)^{-1}$	Overall** Fouling Resistance $(BTU/Hr/Ft^2/°F)^{-1}$	Heat Duty BTU/Hr
3-9-46					Unit Started Up After Test and Inspection							
3-13-46	Chg.Gas Oil	Slurry	1130	1778	135 → 443	458 ← 612	228	73.6	.00418	.00985	0	60,000,0000
3-22-46	Chg.Gas Oil	Slurry	1145	1635	138 → 430	490 ← 648	272	59.4	.00417	.00965	.00303	57,600,000
4-1-46	Chg.Gas Oil	Slurry	888	1635	115 → 416	505 ← 630	262	48.4	.0051	.00965	.00590	45,200,000
4-14-46	Chg.Gas Oil	Slurry	1339	1632	140 → 346	535 ← 657	348	36.6	.00416	.00825	.01494	45,500,000
4-15-46					Caustic Washed							
4-18-46	Chg.Gas Oil	Slurry	1195	1561	140 → 401	495 ← 648	290	50.7	.00430	.01005	.00538	52,550,000
5-3-46	Chg.Gas Oil	Slurry	1105	1530	130 → 400	496 ← 642	294	47.5	.00445	.01005	.00652	49,950,000
5-12-46	Chg.Gas Oil	Slurry	1080	1530	128 → 388	505 ← 635	302	42.4	.00448	.01005	.00907	45,750,000
5-20-46	Chg.Gas Oil	Slurry	1120	1490	135 → 382	512 ← 649	314	40.0	.00500	.00985	.01015	44,950,000
5-22-46					Caustic Washed							



Tube Passes	2	
Shell Passes	1	
Tubes: No	175	
Size	1-1/4"x15.6	
Gauge	10	
Pitch	1-9/16"□	
Outside Area	893	1 2 3 4
Baffle Space	5-3/4"	
Installation Date		3-3-44
Material		Steel

**Includes Tube Thermal Resistance of .00045 $(BTU/Hr/Ft^2/°F)^{-1}$

Flow Materials Properties

	Gas Oil	Catalyst Slurry
Avg. Gravity, °API	30.3	19.0
Avg.Viscosity,Centipoises*	1.1	1.0

*At Average Film Temperatures of:

Gas Oil	237°F.
Slurry	469°F.

ng methods are high, but it is usually necessary to employ these nethods in order to obtain satisfactory results. We have found imited application for cleaning the shell side of heat exchangers y caustic-washing. The gas oil-catalyst slurry exchangers are cleaned by this method. The use of caustic solution in maintaining clean gas-oil preheaters without pulling the bundles has resulted in decreased idle unit time for cleaning and inspection, and at the same time a higher level of heat transfer.

Fouling of Marine-Type Heat Exchangers

By H. E. Bethon,[1] Washington, D. C.

The paper emphasizes the importance of making due allowance for fouling and cleaning in the initial design of heat exchangers for naval service. It describes briefly the design and construction of marine-type heat exchangers (lubricating-oil and jacket-water coolers, steam condensers, steam- and thermal-compression distilling plants and fuel-oil heaters), in addition to various design and operating procedures which affect fouling characteristics.

Introduction

In order that the maximum striking or defensive power of a task force can be realized during a naval assignment, it is important that the design, construction, operation, and maintenance of major machinery components aboard naval vessels be standardized to the greatest extent possible so that each machinery component can be assured of operating at its peak efficiency. In other words, if there were no high degree of uniformity then certain machinery components, having different fouling characteristics, would be forced to shut down for cleaning or operate at reduced power and/or lowered efficiency. Toward this end the practice of the Bureau of Ships is to submit new designs of heat exchangers proposed by industry to governmental laboratories for the purpose of conducting performance tests in addition to evaluating fouling, cleaning, repair, and maintenance characteristics under conditions simulating actual shipboard operation before final approval is granted.

Heat-exchanger operation and maintenance aboard a naval vessel presents an ever-changing picture because, dependent on the vessel's operation in waters varying between the arctic and tropical regions, aside from operation in contaminated rivers and harbors, different marine growths and organisms and variations in chemical constituents are experienced, with the result that cleaning and operating methods must be varied to suit particular circumstances. Furthermore, naval vessels arriving in foreign ports of call often encounter different inferior grades of fuel and lubricating oils that adversely affect the performance of the related heat exchangers.

As regards heat-exchanger design, it safely can be assumed that there is a relatively fair degree of uniformity among manufacturers in the establishment of clean-tube heat-transfer rates as the result of the efforts toward standardization on the part of the Heat Exchange Institute and Tubular Exchanger Manufacturers' Association, among others. These rates are usually determined from velocity versus heat-transfer-rate curves adjusted for viscosity and temperature plus other physical variations, e.g., baffle type and pitch, tube length, diameter, and material, etc., in the case of simple installations such as steam condensers, feedwater heaters, fuel-oil heaters, etc. In the case of complex installations wherein the temperatures of both fluids being handled vary, such as liquid-to-liquid exchangers, gas-to-liquid exchangers, gas-to-gas exchangers, etc., it is customary to employ the resistance method wherein the over-all clean-tube heat-transfer rate is equal to the reciprocal of the sum of the film resistance on the shell side plus the film resistance on the tube side plus the metal resistance.

However, very little has been done to establish a standard method for determining fouling resistances normal to the operation of the heat exchanger, with the result that a manufacturer may offer a design which does not allow adequately for a reasonable degree of fouling, either through lack of knowledge as to the fouling characteristics of the fluids being handled or to a strong endeavor to secure the contract. The result is that the purchaser finds it necessary to resort to frequent shutdowns of the plant for cleaning operations or to continue operation at reduced capacity and/or lowered efficiency. The initial design of a heat exchanger to meet the rated or designed capacity under the specified temperature and pressure conditions is important, but the ultimate successful operation of the unit is dependent primarily on the factors allowed for fouling and cleaning.

The heat exchanger should be designed so that the rated capacity will be realized after a specified number of hours under normal operating conditions has elapsed, thereby directly reflecting the period of satisfactory performance before cleaning becomes necessary. Under all circumstances the heat-exchanger manufacturer should indicate both the clean-tube and over-all heat-transfer rates. The Bureau of Ships has for years adopted this procedure in connection with distilling plants whereby the rated capacity of the plant is understood to be the output of the plant after 30 days of continuous operation, with clean-tube capacity arbitrarily established at 30 per cent in excess of rated capacity. Fouling-resistance rates are a function of the characteristics of the fluids being handled, velocity, tube material, temperatures, tube size, viscosity, among other items. Other factors besides fouling due to normal operation that must be considered in arriving at the optimum heat-exchanger design are weight, space, man power, time outages, load factor, etc. For example, in the case of a submarine, where weight and space are at a premium, it is desirable to design a heat exchanger with reduced allowance for fouling, with consequent frequent cleaning periods, in order to keep weight and space at a minimum, whereas in the case of larger vessels where weight and space are not as critical, a greater allowance can be made for fouling, with the result that prolonged operation at rated capacity can be expected before cleaning becomes necessary.

Design Considerations

In order to establish the proper design of heat exchanger which will be capable of sustaining the required capacity for the desired period of operation between cleaning periods with minimum maintenance and repair, the designer must give serious consideration to the selection of materials which offer maximum resistance to corrosion, to the fouling characteristics of the fluids being handled, and to the provision of adequate means for either manual or chemical cleaning of the unit when found necessary. In the case of heat exchangers, whose performance is greatly affected by fouling, the purchaser should be given as much information as possible on the proper chemical and/or mechanical methods to be employed in the cleaning of these units, in order that their maximum useful service life may be realized. By the same token the purchaser should acquaint the manufacturer fully with information as to the approximate period of storage under atmospheric conditions before installation, in order that the proper allowance for atmospheric corrosion can be made; as to conditions existing

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Note: The opinions expressed herein are those of the author and do not necessarily represent the opinions of the Navy Department or the service at large, nor of the Society. Paper No. 48—A-145.

during installation that may result in unsuspected internal fouling of the unit from an outside source which, if serious enough, may justify the adoption of a removable tube-bundle design or special cleaning features; as to possible abnormal sources of fouling or contamination not expected during normal operation of the unit which may be present, such as, seaweed, kelp, fish, barnacles, and the like. Other circumstances, not ordinarily apparent to the manufacturer, which may have a strong bearing in determining the proper design of heat exchanger should be noted, particularly those items which will result in prolonged satisfactory service life with minimum maintenance, repair, and overhaul.

Causes of Fouling

Fouling of a heat exchanger may occur from the following causes among others:

(*a*) Fouling of transfer surfaces resulting from normal operation, such as deposition of scale, accumulation of slime, sludge, carbon deposits, etc.

(*b*) Fouling of transfer surfaces resulting from internal corrosion of ferrous subcomponents such as baffles, tubes, shell interiors, etc.

(*c*) Fouling of transfer surfaces resulting from foreign matter being carried into the heat exchanger by the circulating sea water, fuel oil, etc., from carelessness during the installation period, and from inadequate methods of cleaning the system before operation is started.

Design Factors That Minimize Fouling Problem

Aside from the importance of designing a heat exchanger with proper attention being paid to adequate allowance for fouling, selection of suitable materials, and to effective cleaning methods, every consideration should be given to establishing basic design requirements which will minimize the fouling or scaling problems. The following examples are cited:

(*a*) Design of distilling plants on the basis of vacuum operation wherein the last-effect shell vacuum is specified at 26.5 in. of mercury, and the desuperheated steam to the first effect is maintained at subatmospheric pressures through the use of an orifice in the steam line. In this manner the temperatures throughout the system are held at low levels, thereby reducing scale formation.

(*b*) Adoption of satisfactory feed-treatment systems, such as the introduction of boiler compound and cornstarch, with the result that the formation of scale is greatly retarded, and the nature of the scale deposit is radically changed from a type which adheres rigidly to the tube surface and defies removal, to a type which is soft and powdery whose removal can be easily accomplished by means of a hosing-down operation. Current investigations are being undertaken on other methods of evaporator feed treatment such as ion exchange of the self-regenerating type, carbon-dioxide removal, electrolytic-cell installations, introduction of different types of chemicals, etc. It must be determined carefully beforehand that these feed-treatment systems will not result in accelerated corrosion or deterioration of the heat exchanger, or adversely influence the successful operation of the plant.

(*c*) Installation of a fresh-water-cooled lubricating-oil system for Diesel engines whereby the engine-jacket water, which has been treated by the introduction of an inhibitor, is employed to cool the engine lubricating oil, while the jacket water in turn is cooled by sea water. This arrangement requires only one sea-water-operated heat exchanger, thereby reducing the fouling and maintenance charges.

(*d*) Introduction of chlorine or chlorine compounds, the provision of a chlorine generating cell in sea-water piping systems resorting to the periodic flushing of piping systems, including h exchangers with heated sea water will reduce the contaminati from marine growths and organisms.

(*e*) The use of low-viscosity high-grade fuel oils will redu the operating temperatures necessary for proper atomizati with the result that less fouling of fuel-oil heaters is encounter

This paper deals with fouling problems in connection with t following types of heat exchangers installed aboard na vessels: Lubricating-oil and jacket-water coolers, steam co densers, fuel-oil heaters, and distilling plants.

LUBRICATING-OIL AND JACKET-WATER COOLERS

Construction of Coolers

All main- and auxiliary-turbine lubricating-oil coolers install aboard surface vessels are of the shell-and-tube type, better kno as type A, with the lubricating oil circulated through the she and with the sea water circulated through the tubes. The ma turbine coolers, varying in size between 300 and 1600 sq ft, are the fixed tube-sheet design, as illustrated in Fig. 1 and are co structed of $^5/_8$-in-OD 70:30 copper-nickel tubes, and 70 copper-nickel tube sheets, with cast bronze water boxes and wi steel shell and baffles, wherein the baffles are either welded bolted to the interior of the shell. Owing to difficulties expe enced with the corrosion of the steel shell and baffles and, since was impossible to inspect and/or clean these parts without tubing the unit completely, the Bureau of Ships, in 1944, requir that the shell and baffles be made of nonferrous material and th the baffles be supported by tie rods and spacers attached to o tube sheet. Thus the entire baffle assembly could be wit drawn after the tubes were removed, for the purpose of cleani the shell interior. Recently, for new construction, the specific tions for main-turbine oil coolers have been changed to require withdrawable tube-bundle design, based upon an outside pack floating tube sheet provided with telltale openings so as to dete immediately either a lubricating-oil or a sea-water leak. T auxiliary-turbine coolers, being much smaller in size, are eith of the fixed or floating tube-sheet design and are usually co structed of $^3/_8$-in-OD 70:30 copper-nickel tubes and with no ferrous shell and baffles.

The main and auxiliary Diesel-engine lubricating-oil a jacket-water coolers installed aboard all submarines and aboa the majority of Diesel-driven surface craft, one notable excepti being LST's, are known as type B units with the lubricating or jacket water circulated through the tubes, and with the s water circulated through the shell. These units are manufactur by the Harrison Radiator Division of General Motors Corpo tion and are furnished in many different sizes and models eith of two basic groups, the largest unit being equivalent to a she and-tube unit containing approximately 400 sq ft of cooling su face. The two basic groups are plate-tube type and strut-tu type. Fig. 2 illustrates the general principles of construction o Harrison strut-tube-type cooler for an emergency Dies engine generator set.

The plate-tube-type cooler consists of a stack of plate-ty tubes connected in parallel with the oil supply and enclosed in cast-bronze housing through which sea water is circulated. this design the plate assembly is removable. The fabricat strut-tube-type cooler comprises two different designs of str tubes, one for water cooling and the other for oil cooling. T water tube is from 2 to 3 in. wide and is composed of a male a female section, each containing a series of formed dimples, whi are copper-brazed together in a hydrogen furnace. The oil tu is 1 in. wide and is fabricated in the same manner as the plat type tube, wherein the male and female sections, which enclo

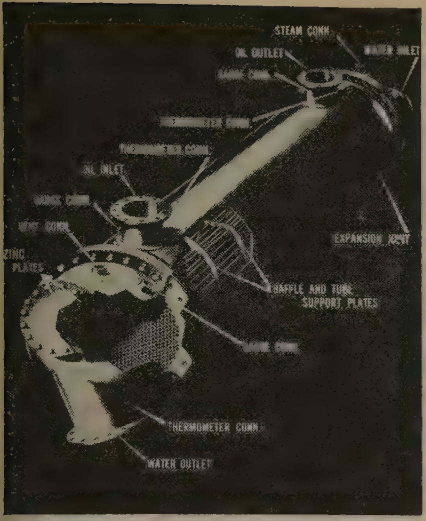


FIG. 1 MAIN-TURBINE LUBRICATING-OIL COOLER, ALCO PRODUCTS

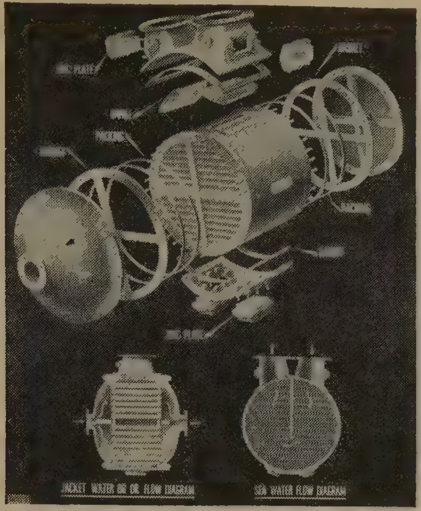


FIG. 2 HARRISON LUBE-OIL AND JACKET-WATER COOLERS

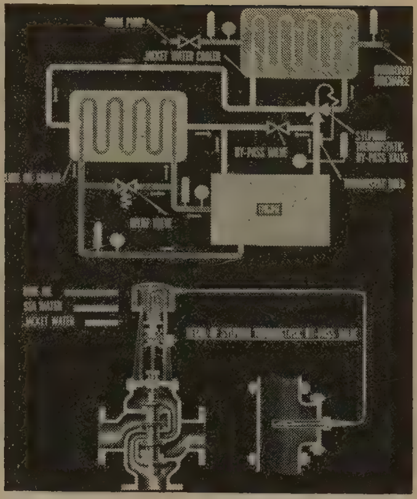


FIG. 3 LUBE-OIL AND JACKET-WATER COOLING SYSTEM

an internal distributor strip, are copper-brazed together in a hydrogen furnace.

The material of the plate- and strut-type tubes is 70:30 copper-nickel. The plate-type tubes are copper-brazed to a 70:30 copper-nickel header plate as an entire assembly. The strut-type tubes are soldered into a 70:30 copper-nickel header plate which in turn is soldered to the frame of the cast-bronze casing. All strut-tube-type models except Model HE24 are of the fixed tube-bundle design. In the case of new construction, the shell or casing and oil or jacket-water heads or covers will be of brazed copper-nickel plate construction in lieu of castings, whereas the strut-type tubes will be silver-brazed in lieu of soldered to the header plates, and the tube-bundle assembly will be of the removable type.

Piping Arrangement

Diesel-engine installations aboard submarines and surface vessels are based upon the lubricating-oil and jacket-water cooling system as illustrated in Fig. 3, wherein the lubricating oil is cooled by the engine jacket water which is in turn cooled by sea water. In this system sufficient by-passes and thermostatic controls are provided for the purpose of maintaining optimum lubricating-oil and jacket-water temperatures, regardless of engine loads and injection temperatures.

Performance of Type A Coolers

The following cases are cited to illustrate the seriousness of fouled main-turbine lubricating-oil coolers of the fixed tube sheet and baffle design installed aboard various types of naval vessels which occasioned a large amount of additional work that would not have been necessary had these coolers been of a design which permitted access to clean them without retubing, or had they been constructed of noncorrosive material throughout.

U.S.S. Shaw (DD373). This vessel was damaged during the Pearl Harbor attack on Dec. 7, 1941, and the engine room flooded. The vessel was later run aground. Among other spaces, the engine room was flooded for the second time. Piping was thoroughly cleaned and lubricating-oil coolers were sent to the shop, cleaned with trichloroethylene solution, and tested afterward only hydrostatically. On completion of machinery work, the system was flushed thoroughly with oil several times, and the bearings were inspected and found satisfactory. After 6-hr full-power operation at sea, the lubricating-oil pumps failed on the port unit. On arrival at the yard with the port unit secured, the bearings in the port engine were inspected and found filled with abrasive rust, and many journals scored and bearings wiped. All upper gears were removed and sent to the shop for regrinding and resizing to common dimension; all bearings were remetaled. The lubricating-oil coolers were taken to the shop with instructions to open up and retube. When the tubes were cut out and coolers opened, excessive corrosion was found inside the wrapper sheet and on the baffles. The coolers were pickled thoroughly and baffles replaced.

This casualty was caused by the corrosion existing inside the lubricating-oil coolers, which had not been eliminated by cleaning with trichloroethylene. This corrosion was occasioned by the two previous floodings of the engine-room spaces, wherein sea water may have been introduced into the coolers. Cleaning with trichloroethylene did not remove this old rust. Fig. 4 shows the exterior of the aft-end tube bundle of the port lubricating-oil cooler. The baffles in the shell were badly corroded and coated with iron-oxide grit.

U.S.S. Pensacola (CA24). Due to damage and submersion in sea water, all machinery units, piping, and lubricating-oil coolers in the after engine room were removed for cleaning and repairs. The two lubricating-oil coolers were completely disassembled, shells cleaned, tube sheets cleaned, and then the coolers were retubed. The coolers were reinstalled on board ship in March, 1943. About September 1, 1943, an examination was made of the oil and water sides of all coolers. Coolers in the forward engine room were found to contain much sludge and grit in the oil side of the shell. Coolers in the after engine room were found to contain rust and fine grit in sufficient quantity to warrant recleaning after having been installed about 5 months.

All coolers were taken to the shop, tubes removed, shells cleaned, reassembled, and retubed. After retubing, all coolers were tested to 150 psi with oil. All oil piping in both engine rooms was taken to the shop, cleaned, annealed, and reinstalled, followed by an extensive flushing schedule. Fig. 5 shows the exterior of the tube bundle of the cooler in the after engine room. The baffles were badly corroded and heavily coated with iron-oxide grit.

Numerous similar instances can be cited wherein main-turbine oil coolers were fouled to the extent of rendering them inoperative. Internal corrosion of the steel shell and baffles was caused by atmospheric conditions during prolonged periods in storage

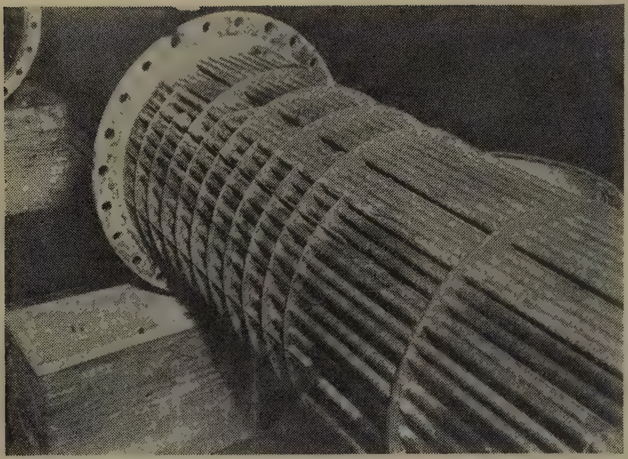


Fig. 4 Tube Bundle of Lube-Oil Cooler—Destroyer U.S.S. "Shaw" (DD373)

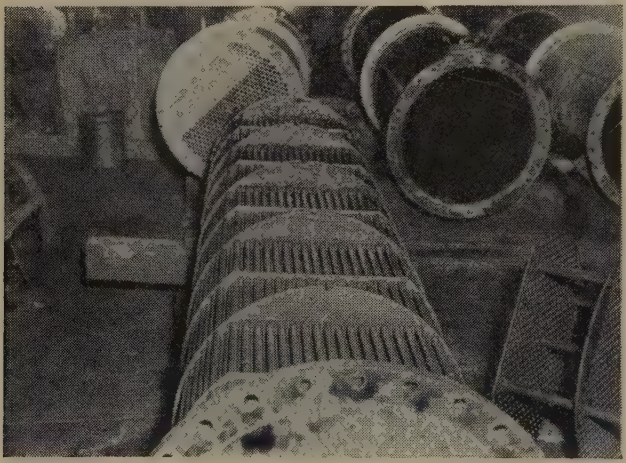


Fig. 5 Tube Bundle of Lube-Oil Cooler—Cruiser U.S.S. "Pensacola" (CA24)

prior to installation aboard ship. Fouling was also caused by the lodgment of foreign matter such as rust, rags, weld beads, excessive dirt, chip drillings, etc., in the shell of the oil cooler, resulting from careless installation procedures. In some extreme cases the cooler tubes had to be removed completely in order to provide access to the shell side for cleaning purposes.

Performance of Type B Coolers

In the case of type B (Harrison) lubricating-oil and jacket-water coolers wherein the flow passages are exceedingly small (less than $^1/_8$ in.), strainers having a fine-mesh screen are usually provided in the sea-water circuits ahead of the cooler. No unusual fouling difficulties, except for rust scale, have been experienced in the lubricating-oil and jacket-water sides of these coolers during normal operation inasmuch as any foreign matter, the entry of which occurred during the ship construction period, will be deposited at the entrance to the tubes owing to the small flow passages, the removal of which can be readily accomplished.

In May, 1947, three Model 27 Harrison lubricating-oil coolers serving one main-turbine unit of the cruiser U.S.S. *Rochester* (CA124) completed over 4300 operating hours of satisfactory service performance without any leakage or fouling difficulties being encountered throughout the entire period. The three units were arranged in parallel on the oil side and in series on the sea-water side. No strainers were installed in the cooling-water system, and these units were not given any more operating attentio than the standard shell and tube unit serving the adjacent mai turbine. The great savings in weight and space of the Harriso units over the standard $^5/_8$-in-OD tube unit was particularl noticeable.

Just recently, the Harrison units were removed from th U.S.S. *Rochester* (CA124) and in their place has been installe one Ross $^1/_4$-in-OD tube cooler, provided with a removable tub bundle, for the purpose of observing the fouling characteristic of this unit under actual operating conditions. This cooler, alon with a $^3/_8$-in-OD tube cooler of the same design and construction, recently passed performance tests at the Engineering Experiment Station, Annapolis, Md.

Fouling of Coolers

The fouling of heat exchangers aboard naval vessels depend to a large extent on the sea-water characteristics in the region c the vessel's operations. Reports of considerable fouling of Ha rison type and of small shell-and-tube-type heat exchange serving auxiliary machinery units have been received from th destroyer tenders U.S.S. *Dixie* (AD14) and U.S.S. *Piedmo* (AD17). Approximately 300 hr of operation were obtainab between cleanings.

The warm and highly polluted waters of the operating are contributed to the rapid growth of fouling agents, which consis primarily of hydroids, shellfish (mussels), algae, and a moss substance similar to fine grass roots. In both cases, strainers c sufficiently fine mesh are installed ahead of the heat exchanger Fouling apparently is caused by embryo shellfish passing throug the strainers and adhering to the interiors of the heat exchanger Within 12 days after entry, the embryo fish grow to such propo tions as to cause complete disruption of the flow of cooling wate with the resultant effect that in some cases machinery units hav been rendered inoperative until cleared of the fouled condition.

The cleaning methods employed in the cases cited are strictl mechanized and manual. Low-pressure steam was employed t dislodge the growths and clean the surfaces down to bare meta A diver was also sent overside periodically to remove the mass c marine growth from the sea strainers. The fouling is not excessive when there is a reasonably rapid flow of water through th heat exchanger.

In general, submarines have reported practically no foulin and very little scale on the sea-water sides of heat exchange even after long hours in use; several instances have exceede 5000 hr with the result that these units yielded their full capacit output until they failed in service. It is believed that chemic action during acid-cleaning performed at a naval shipyard wa responsible for their failure.

The submarine tender U.S.S. *Orion* (AS18) reported that th submarines under her care operated as much as 5000 hr withou cleaning and, except in the case of one ship which operated in th Mississippi River, no unusual operating conditions were er countered. Operations in the river caused mud to be deposit on the cooling surfaces, sufficient in some cases to cause comple stoppage. Main motors and main engines overheated at norm cruising speed on two engines. The major part of the mud fou ing was cleared by submerging to 300 ft and opening the cool drains and later by blowing through with air at 225 psi.

The submarine tender U.S.S. *Nereus* (AS17) reported th upon removal of the eight main-engine jacket-water coolers an of the eight main-engine lubricating-oil coolers, both of which ar of the shell-and-tube design ($^3/_8$-in-OD), for cleaning and testi after the main engines operated an average of 2055 hr each, a proximately 20 per cent of the tubes were found to be plugge and the interior of all tubes was coated with a white hard sca averaging about $^1/_{64}$ in. in thickness. The zincs were inspected and cleaned monthly and renewed when 50 per cent deteriorated. In addition, a considerable amount of marine growth and sediment was also present, despite the fact that strainers are installed in the sea-water lines ahead of all heat exchangers. Low and high sea-suction lines were found to be badly fouled throughout with marine growth. No ill effects or abnormal temperatures were noted due to the 20 per cent plugging of tubes or marine growth found in the heat exchangers. During the same overhaul period, the four auxiliary-engine jacket-water coolers and the four auxiliary-engine lubricating-oil coolers were removed for inspection. These units are of the Harrison design. The zincs were inspected and cleaned weekly and renewed when 50 per cent deteriorated. No unusual operating conditions existed other than the use of harbor water for protracted periods while at anchor. Upon inspections, these coolers showed small amounts of marine growth and sediment; no scaling was in evidence. There have been no ill effects or abnormal temperatures experienced due to scaling or marine growth.

Operation of Coolers

Particular care must be taken in the operation of lubricating-oil coolers to avoid chilling the lubricating oil, which may result from full flow of cooling water and from by-passing the lubricating oil, in order to eliminate sludge precipitation which reduces the heat-transfer rate, thus necessitating frequent cleanings. Furthermore, too low an oil temperature (below 130 F) may result in getting condensation into the bearing housings and gear casings. This entry of water into the lubricating-oil system sooner or later will prove troublesome. The cruiser U.S.S. *Topeka* (CL67) reported the failure of worm-gear casings on the condensate and booster-pump units due to operating with too cold an oil temperature.

In order to minimize the fouling and corrosion of the sea-water side of the Harrison type jacket-water coolers installed aboard the LSM's (landing ships, medium), it was found desirable to provide built-in air-separating tanks, connect the salt-water side of the main-engine fresh-water coolers in parallel, coat the interiors of the sand traps and air-separating tanks and all surfaces of the strainer screens in the sand traps with antifouling paint, and install water-actuated sand eductors in the sand traps among other corrective measures.

Fig. 6 illustrates the extent to which a small marine Diesel-engine water cooler was plugged with lint and rags on the water side in the vicinity of the zinc pencil, which probably was caused by careless installation procedures.

Fig. 7 illustrates the fouling of a Model HE424-362 Harrison strut-tube-type lubricating-oil cooler removed from the seaplane tender U.S.S. *Barnegat* (AVP10) after being in service over 3 years. The fouling on the inlet-water side with stones, mud, and silt can be seen plainly.

Fig. 8 illustrates the fouling of a Model 60 Harrison plate-tube-type lubricating-oil cooler also removed from the seaplane tender U.S.S. *Barnegat* (AVP10) after being in service over three years. The fouling of the plate tubes on the water side with marine growth can be observed readily.

Fig. 9 illustrates the fouling on the lubricating-oil side with carbon sludge of a Model HE124-250 Harrison strut-tube-type lubricating-oil cooler removed from the tug *Sheila Moran* after being in service approximately 15 months.

Steam Condensers

Construction of Condensers

The main condensers installed aboard combatant naval vessels and the majority of large turbine-driven auxiliaries are of single-

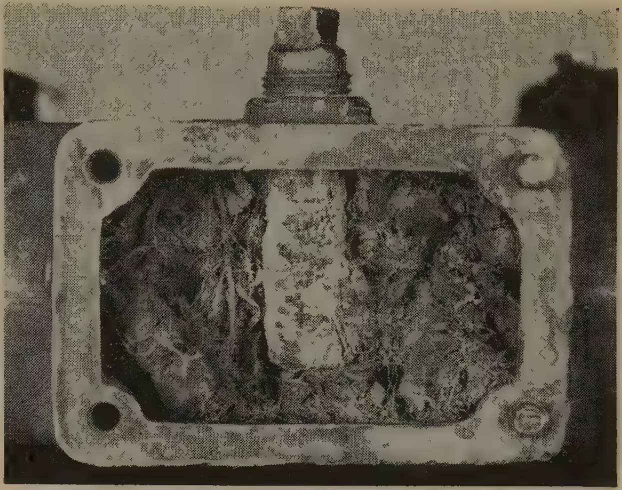


Fig. 6 Fouled Condition of Small Harrison Diesel-Engine Water Cooler

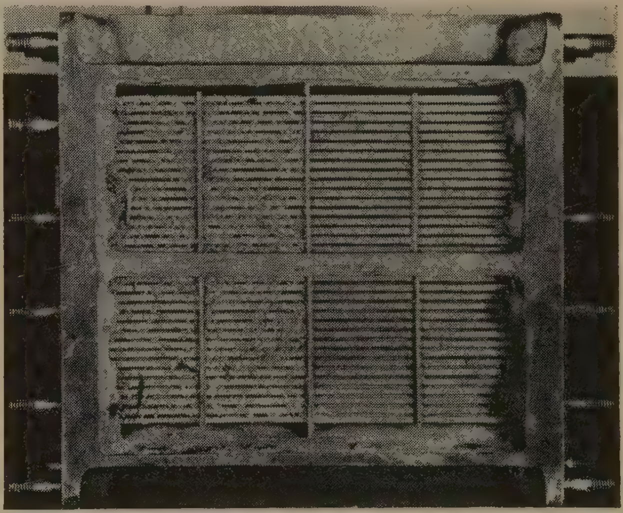


Fig. 7 Fouled Condition on Water Side of Harrison Model HE424-362 Oil Cooler

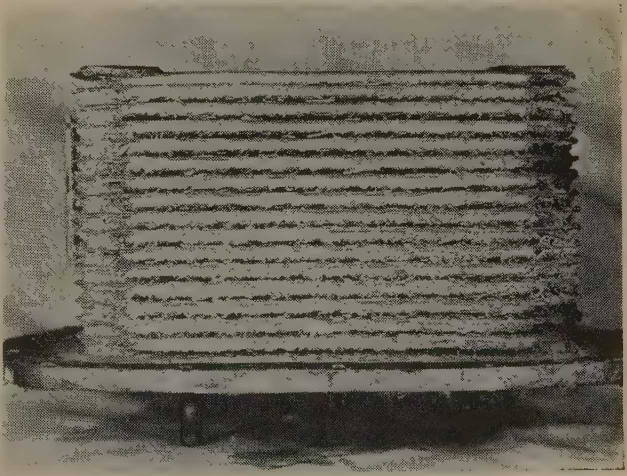


Fig. 8 Fouled Condition on Water Side of Harrison Model 60 Oil Cooler

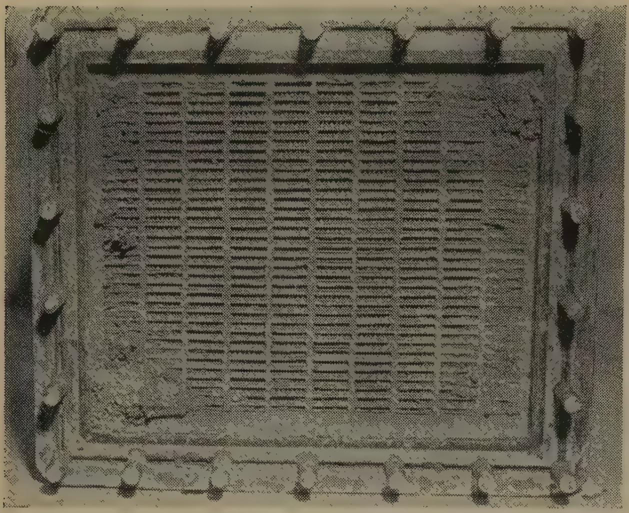


Fig. 9 Fouled Condition on Oil Side of Harrison Model HE124-250 Oil Cooler

pass construction employing $^5/_8$-in-OD copper-nickel tubes wherein the circulating water is provided by a scoop injection system. A circulating pump, taking suction from a separate sea chest, is provided for low speeds and for astern operation. The main condensers installed aboard vessels built to the specifications of the U. S. Maritime Commission which were later converted to naval auxiliaries are, in general, of two-pass construction employing $^3/_4$-in-OD aluminum-brass tubes, in which the circulating water is provided by a circulating pump. The main condensers vary in size from 3,800 sq ft to 28,000 sq ft, the latter size being installed aboard the CVB41 class aircraft carriers.

Fig. 10 illustrates the main condenser provided for the BB61 class battleships The water boxes are of nickel-copper (monel) welded-plate construction of the sectional type, the tubes and tube sheets are of (70:30) copper-nickel material, whereas the shell-and-tube support plates are of steel plate. The main condensers are installed either athwartship or fore and aft dependent upon whichever location bests suits the arrangement of machinery in the engine rooms. The dynamo or auxiliary condensers are of two-pass construction employing the same materials as in the case of the main condensers, except that the water boxes are of cast bronze in lieu of monel metal. Fig. 11 illustrates the auxiliary condenser provided for the DD445 class destroyers.

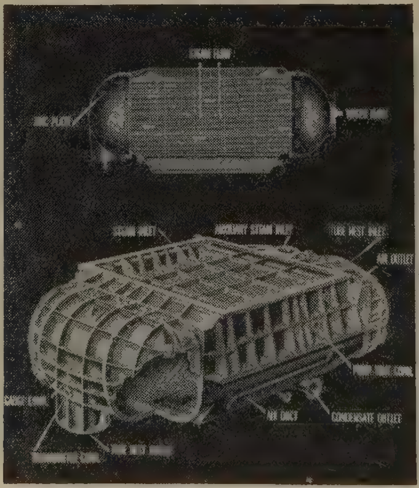


Fig. 10 Main Condenser—BB61 Class Battleships, Foster-Wheeler

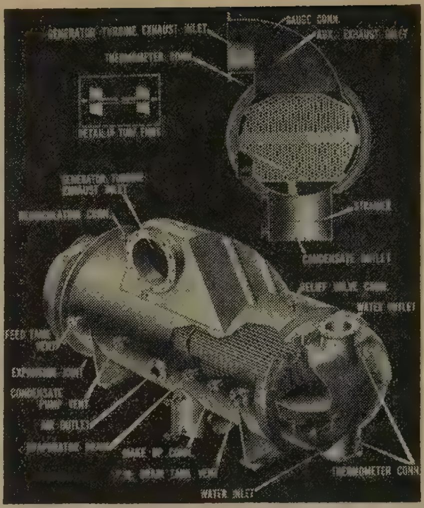


Fig. 11 Auxiliary Condenser—DD445 Class Destroyers, Foster Wheeler

Fouling of Steam Side

Ordinarily the fouling of the steam side of the condenser under normal operating conditions presents no undue difficulties. The removal of any grease or dirt is periodically accomplished by boiling out with a strong solution of Navy standard boiler compound at intervals of every 2 or 3 years. In order to eliminate atmospheric corrosion of the shell interiors, the Bureau of Ships, in 1944, required that a coating of corn oil and Japan drier or an equivalent protective coating be applied to the inside of the steel-plate shell and to the steel tube-support plates. Under these circumstances it is possible to maintain condensers satisfactory in open storage for extended periods without incurring any serious corrosion of the condenser shell. Fig. 12 illustrates the foreign materials removed from No. 1 dynamo condenser aboard the U.S.S. *Wichita* (CA45) resulting from a casualty to No. 1 turbogenerator. The foreign materials comprise primarily pieces of turbine blading and of carbon packing ring.

Fouling of Water Side

On the other hand, it is extremely important to maintain the inside of the condenser tubes free from fouling inasmuch as the service life of the condenser tubes is greatly dependent thereon.

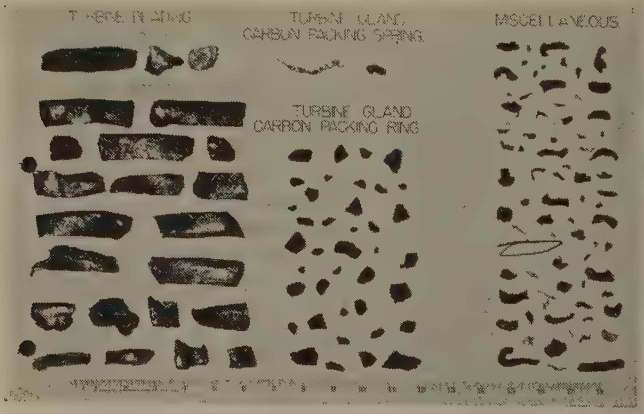

TURBINE BLADING
TURBINE GLAND CARBON PACKING SPRING
TURBINE GLAND CARBON PACKING RING
MISCELLANEOUS

Fig. 12 Foreign Materials Removed From No. 1 Dynamo Condenser U.S.S. "Wichita" (CA45)

Under normal operating conditions, the fouling is not severe enough to affect seriously the operating efficiency of the prime mover but usually results in accelerated tube failures which require the turbine to be secured for the purpose of plugging or replacing the leaky tubes. Owing to the low circulating-water temperatures and to the favorable scaling characteristics of copper-nickel tubing, the amount of scale formed on the inside surface of the condenser tubes is practically nil. Hence the fouling of condenser tubes is occasioned mainly by the entry of foreign matter entrained with the incoming circulating water. The lodgment of foreign matter in condenser tubes causes two types of tube failure, dependent upon the size of the foreign particles entering the tubes. A foreign particle large enough to have appreciably obstructed the flow of water through the tube may cause failure at the point of contact due to corrosion, or the failure may occur in the area just downstream from the point of lodgment due to erosion. Lodgment of small foreign particles usually takes place when the condenser is being operated at a very low rate of sea-water circulation or when an idle condenser is allowed to remain flooded with polluted harbor or river water.

The draining of the sea-water side of an idle condenser may prove more dangerous than leaving the condenser flooded with polluted water, unless care is exercised that the condenser is properly cleaned at the time of draining and the tubes left dry. In the event the condenser is drained and allowed to stand without cleaning, polluted water in low points along the tube will evaporate gradually, concentrating the impurities which are left behind as the evaporation proceeds. Thus highly polluted water, concentrated in small areas along the bottom of the tubes, corrodes and grooves the tube wall in these areas. When condensers are drained in port, it is highly desirable that the sea-water side be washed out thoroughly with fresh water. If ample fresh water is available, the most desirable method is to water-lance each tube, using fresh water.

During the summer of 1944, the Philadelphia Naval Shipyard was very much concerned with the serious water conditions in the Delaware River resulting from the generation of hydrogen sulphide fumes arising from the decomposition of untreated sewage and waste. As a result of this contaminated water condition, the Bureau of Ships issued separate instructions requiring the inspection and cleaning of condenser tubes in the case of vessels proceeding to sea from ports such as Philadelphia where several hours of operation in a relatively shallow channel through polluted water is necessary. The cleaning is to be undertaken at the first available opportunity occurring after the vessel reaches clear water.

Examples of Condenser Fouling

Numerous cases are on file wherein main-condenser tubes have been fouled by marine life such as fish, eels, "Japanese worms," etc., to the extent that the condenser vacuum suddenly dropped to a point at which the efficiency of the main turbine was adversely affected. The main-injection sea chest is not provided with a strainer but is provided with strainer bars, spaced about 2–3 in. apart, with the result that marine life enters the main injection system and fouls up the condenser-tube entrances. Fig. 13 illustrates a severe case of fouling at the inlet tube sheet of No. 1 main condenser installed aboard the aircraft carrier U.S.S. *Saipan* (CVL48). This view was taken on December 3, 1947, and judging from the diameter of the belled ends of the plastic condenser-tube inserts, it can be seen readily that these fish are on the average about 10–12 in. in length.

The destroyer U.S.S. *Fullam* (DD474), on June 17, 1945, reported that after completing 78 days' steady steaming in the Ryukus Area the No. 1 main condenser was drained and opened and thousands of living, white squirming "worms" were found sticking to the sides of the water box, tube sheets, and, most important, inside the tubes themselves. None of this marine growth was found in an attached condition on the discharge side. It is estimated that these organisms were found in 40 to 50 per cent of the tubes. Samples of this marine growth were forwarded to the Bureau of Ships and were chosen to show progressive stages of growth which varied from infinitesimal white spots to a growth approximately $^3/_8$ in. diam and 3 to 4 in. long. In the advanced stages of growth the parasite begins to resemble a snail; the living end beginning to harden and having whorls similar to a snail. The outstanding point of this growth is the adhesive pad of fibrous material by which the growth attaches itself to the condenser, necessitating that the growth be scraped or peeled from the condenser surfaces. Samples of this growth were later identified as a Japanese barnacle known technically as *Conchoderma Auritum.*

Fig. 13 Fish Trapped at Inlet Tube Sheet of Main Condenser—U.S.S. "Saipan" (CVL-48)

The condenser was first steamed to kill the growth, the surfaces were then cleaned carefully using a light scraper on the water box. A scrubbing brush and fresh water were used on the tube sheets, and then the tubes were steam-lanced, followed by air-lancing from the discharge side. The debris was cleaned, portable blowers were rigged, and the condenser was allowed to set overnight. In the morning the air-lancing was repeated. With the completion of this process, investigation showed less than 5 per cent of the tubes had any obstruction remaining. Instructions were issued to effect the periodic overheating of the condenser circulating water to a temperature of about 5 deg F above the maximum growth-environment temperature, with the intent of killing the barnacle, thus promoting subsequent rotting and washing away of the remains while under way.

During wartime operations, unusual fouling of the main-injection circulating-water systems and of the distilling-plant-inlet sea chests with fuel oil and its resultant by-products were experienced, especially after a naval vessel passed through a heavy oil slick, resulting from the previous sinking of an oil tanker. In the case of one tanker, a tarry substance approximately $^1/_4$ in. thick covered the inlet tube sheet over large areas and consequently blocked the flow of circulating water through the condenser tubes. In the case of distilling plants, the evaporator tube nests became fouled with a heavy coating of oil, and the units had to be boiled out thoroughly before the resumption of normal operation.

FUEL-OIL HEATERS

Construction of Heaters

Since 1938, all naval vessels have been furnished with type B fuel-oil heaters wherein the heating steam passes through the inner tubes which contain external fins, and the fuel oil passes in a counterflow direction along the fins between the outer tubes and the inner tubes. Since the resistance to heat transfer is preponderantly on the oil side, it is therefore economical to provide extended finned surface in order to keep the weight and space at a minimum. Fig. 14 illustrates a sectional G-fin fuel-oil heater manufactured by the Griscom-Russell Company. In this design the inner tube sections are removable for the purpose of replacement or mechanical cleaning. In general, the fuel-oil heating installation is divided between two or more sections operating in parallel so that each section performs at its maximum capacity under varying load demands, thereby reducing unnecessary fouling of the heaters. Before type B fuel-oil heaters are accepted by the Bureau of Ships, they must pass exhaustive tests at the Naval Boiler and Turbine Laboratory to determine their performance and fouling characteristics in addition to weight and space evaluations. Conventional shell-and-tube type fuel-oil heaters, known as type A, are installed aboard naval vessels built prior to 1938, and aboard merchant vessels converted to naval auxiliaries.

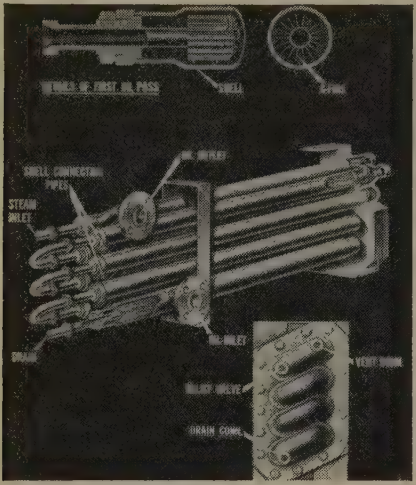


Fig. 14 Sectional G-Fin Fuel-Oil Heater, Griscom-Russell

Fouling of Heaters

The rate of fouling on the oil sides of fuel-oil heaters is governed by three factors, namely, the rate at which oil passes through the heater, the temperature to which the oil is heated, and the nature of the oil itself. Overheating of the oil is probably the greatest single cause of fouling. "Cracked" oils, particularly, break down under heat and deposit carbon residue; such oils are more reduced in viscosity for a particular rise in temperature than straight-run oils. Under these circumstances, it is especially important that cracked oils should not be heated beyond the temperature which gives the correct viscosity for efficient atomization. Care must be taken when heaters are being operated at reduced capacity to insure that excess oil temperatures are not reached. Since the normal rate of operation is low, the fact that heavy fouling exists may not become apparent before an attempt is made to operate the heater at its full rated capacity. To determine the state of fouling during operation at reduced rates, the curves shown in Fig. 15, which were devised by the author in 1935, should be used for predicting fuel-oil-heater performance at full power. When performance checks indicate that a heater has fouled, it should be cleaned at the first available opportunity.

Several destroyers which have just recently returned from operations in the Mediterranean reported a serious fouling condition in the case of Griscom-Russell G-fin type fuel-oil heaters. In each instance much lint had collected on the fins which were visible from the header. Also a thin carbonaceous deposit had collected on all surfaces in contact with the fuel oil. These heaters had not been cleaned since their installation 3 to 4 years ago. The cause of fouling can be attributed directly to the type of oil received in the Mediterranean area. This fuel oil contained much water and insoluble material, some of the latter showed up in the fuel-oil trunk screens in the form of large wads and had to be removed to continue fueling. This oil also had a high viscosity, with the result that it had to be heated to a higher temperature, approximately 180 F. The heaters would not have been so fouled if a strainer had been installed ahead of the unit. The oil strainers located on the discharge side did not become excessively fouled with foreign matter.

Owing to the improved grades of fuel oil employed by the Navy, very little difficulty has been experienced with the fouling of the oil sides of fuel-oil heaters.

DISTILLING PLANTS

Construction of Distilling Plants

In general, distilling plants installed aboard all steam-turbine-driven naval vessels since 1922, have been of the multieffect, low-pressure submerged-tube type, designed to operate on auxiliary exhaust steam at pressures below 5 psi. These plants vary in size from 4, 8, 10, 12, and 20,000 gpd double-effect units to 20, 30, and 40,000 gpd triple-effect units. In the case of Diesel-driven surface vessels and submarines, motor-driven vapor-compression distilling plants, in capacities of 1000 and 2000 gpd have been installed, except for a small number of Diesel-driven auxiliary vessels which are provided with low-pressure distilling plants in which the steam is supplied by small auxiliary steam-generating units.

With the exception of the CV9 and CVB41 class carriers, the majority of low-pressure distilling plants installed aboard naval vessels are of the combined shell construction whereby certain heat exchangers are incorporated within a single-unit assembly. Various different arrangements of distilling-plant combinations have been designed by Griscom-Russell Company, Foster-Wheeler Corporation, New York Shipbuilding Corporation, and Bethlehem Steel Company.

Fig. 16 illustrates a double-effect "solo-shell"-type distilling plant that is furnished in the 4, 8, 10, and 12,000-gpd sizes. The only external unit not shown in this assembly is the condensate cooler. The first- and second-effect evaporators, vapor feed heater, distilling condenser, and vapor baffles and separators are all contained in a single shell with a vertical division plate separating the two effects. The air ejectors, air-ejector condenser, flash chamber, and drain regulators are supported externally by the shell.

Fig. 17 illustrates the 40,000-gpd triple-effect separate-shell distilling plant provided on the CV9 and CVB41 class carriers. The evaporators (3), vapor feed heaters (2), distilling condenser, air-ejector condenser, and condensate cooler are all independent units which are separately piped up to complete the assembly of the distilling plant. The vapor baffles and separators are incorporated within the evaporator shell for each effect.

The material of the evaporator shells is cast bronze or naval rolled brass, except that in the more recent units, welded copper-nickel plate is employed. The material of the tubes and tube sheets in the condensate cooler, distilling condenser, air-ejector condenser, and vapor feed heater(s) is 70:30 copper-nickel. Admiralty tubes and naval brass tube sheets were employed in the construction of the evaporator tube bundles but the Bureau of Ships is now requiring 70:30 copper-nickel for these component parts.

METHOD PROCEDURE

CURVE 'A'

ESTABLISH H.T.R. AS RESULT OF TEST FOR:-

1. RATE OF FLOW
2. SIZE OF TUBES
3. VISCOSITY OF FUEL OIL
4. STEAM PRESSURE

CURVE 'B'

THEN WITH THIS H.T.R. DETERMINE THE FINAL-OUTLET OIL TEMPERATURE FOR:-

1. ANY H.S.
2. ANY QUANTITY OF OIL
3. ANY STEAM TEMP. (SAT.)
4. ANY INLET OIL TEMP.

CONCLUSIONS

1. HEATER SATISFACTORY IF OUTLET TEMP. ON TEST IS ABOVE DETERMINED TEMP.
2. HEATER UNSATISFACTORY IF OUTLET TEMP. ON TEST IS BELOW DETERMINED TEMP.

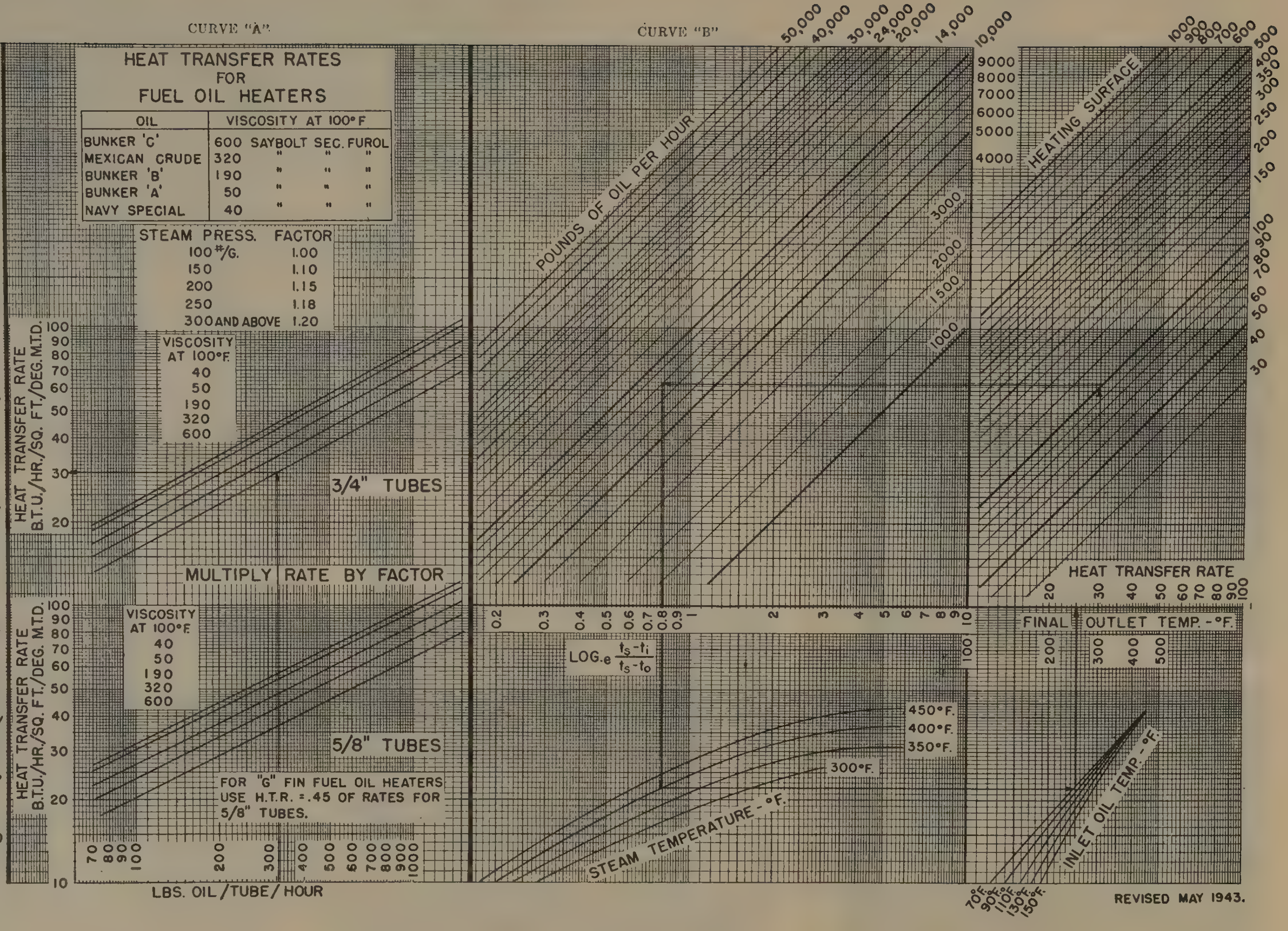
EXAMPLE

CURVE 'A': FOR 325 LB. OIL/TUBE/HOUR; 3/4" TUBE AND 600 S.S.F. OIL VISCOSITY WITH 100# GAGE STEAM PRESS. H.T.R = 30.

CURVE 'B': WITH H.T.R.=30 AND FOR 100 & 8000# OIL/HR., 400° F. STEAM TEMP. AND 70° F. INLET TEMP. FINAL OUTLET TEMP. 249°

CHECK

$$HS = \frac{8000(249-70).48}{30 \cdot 229(MTD)} = 100$$



OIL	VISCOSITY AT 100° F
BUNKER 'C'	600 SAYBOLT SEC. FUROL
MEXICAN CRUDE	320 " " "
BUNKER 'B'	190 " " "
BUNKER 'A'	50 " " "
NAVY SPECIAL	40 " " "

STEAM PRESS.	FACTOR
100#/G.	1.00
150	1.10
200	1.15
250	1.18
300 AND ABOVE	1.20

FOR "G" FIN FUEL OIL HEATERS USE H.T.R. = .45 OF RATES FOR 5/8" TUBES.

Fig. 15 Fuel-Oil-Heater Performance Chart

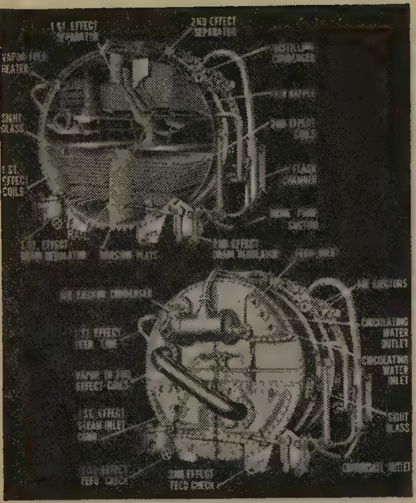


Fig. 16 Double-Effect Soloshell Distilling Plant, Griscom-Russell

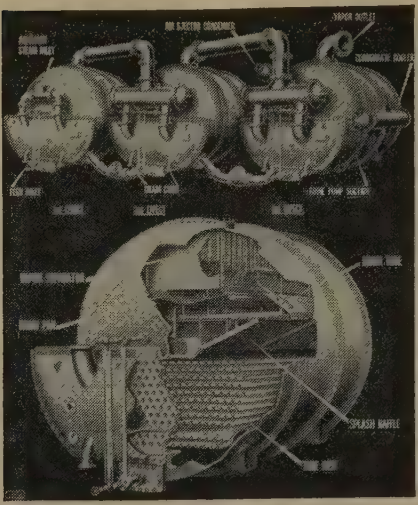


Fig. 17 Triple-Effect Separate Shell 40,000-Gpd Distilling Plant, Griscom-Russell

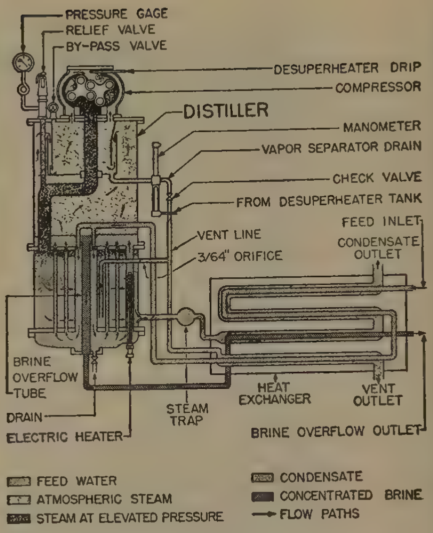


Fig. 19 Badger Model X-1 Vapor-Compression Distilling Unit

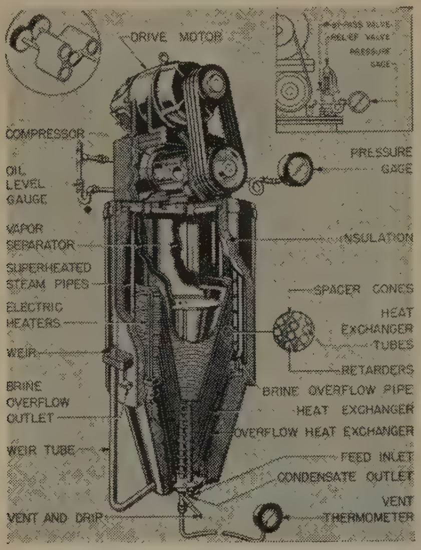


Fig. 18 Kleinschmidt Model S Vapor-Compression Distilling Unit

Fig. 18 illustrates the "Kleinschmidt" model "S" vapor-compression distilling plant which was installed aboard submarines and Diesel-driven surface craft from 1940 to 1943, whereafter the "Badger" model "X" plant, as shown in Fig. 19, was substituted in its place. In the model S plant the evaporation-condensation and feed heating-condensate cooling are accomplished in the same unit. In the model X plant these functions are handled by two independent units known as the evaporator and the heat exchanger. In the case of the model X plant straight-tube construction is used throughout, which permits mechanical cleaning, whereas, in the case of the model S plant chemical cleaning is favored, although the unit can be mechanically cleaned. However, considerable time is required to dismantle and clean the ten conical-shaped coil.assemblies. The material employed throughout the construction of both model S and X plants is 70:30 copper-nickel.

Fouling of Distilling Plants

The principal cause of fouling in distilling plants is the deposition of scale on the tube transfer surface of the evaporators and associated heat exchangers. The rate of scale formation is affected by the operating temperatures, brine density, and nature of the solids in the feed among other factors. In order to keep scale formation at a minimum, the design of the low-pressure steam distilling plant is based on producing the rated capacity with subatmospheric steam pressures (below 212 F), and on maintaining a brine density in the last effect shell not over 1.5 thirty seconds. In addition to these design requirements, the Bureau of Ships has required the installation of a continuous cornstarch boiler compound feed-treatment system. It is recognized that one of the best ways to eliminate or reduce scale formation is through the process of evaporator feed treatment, and numerous research and development projects as hereinafter described have been established toward achieving this goal.

Rear Admiral Thorvald A. Solberg, now Chief of the Office of Naval Research, conceived and initiated the idea of treating the evaporator feed aboard United States naval vessels. During 1941, while he was Force Engineer of the Battle Forces, Pacific Fleet, actual experimental shipboard installations were made. Prior to this period, he conducted numerous laboratory tests on board ship in an attempt to arrive at the most practical method of treating evaporator feed that would obviate the need of carrying enormous supplies of chemicals aboard naval vessels. As a result, a mixture of boiler compound and cornstarch was adopted as the most satisfactory solution for low-pressure-steam distilling plants.

In a recent survey, the Boston Naval Shipyard reported that no unusual scaling was noted in vapor-compression or low-pressure-steam distilling plants. However, it was indicated that certain ships currently in the yard for overhaul have scale of appreciable thickness, about $^1/_4$ in. Other ships operating under similar circumstances and with like periods of cleaning and overhaul of their evaporators, namely, three years, have relatively clean plants. The evaporators on most of the ships mentioned have never been removed and cleaned since their original installation, 3 to 4 years ago. A physical inspection revealed that the scale ranged from $^1/_{32}$ to $^1/_4$ in. thick, the thicker layer being on the shell, which is usually the case. One ship reports that the evaporator capacity was not seriously impaired by this scale. All ships stated that they have followed the shipboard procedure of using cornstarch boiler compound and cold-shocking the tubes for preventive maintenance.

Vessels operating in the Mediterranean area reported that due to the temperature and high carbonate content of the evaporator feed, the tubes become fouled in a very short time. One submarine, after returning from a 108-day patrol in the Pacific area, reported that the distilling plants were operating at near capacity after being in use for about 80 days of the patrol. Since adequate strainers are provided in the circulating-water supply lines of both low-pressure-steam and vapor-compression distilling plants, no unusual conditions of fouling due to entry of foreign matter or marine growth have been experienced.

An excellent argument against the use of ferrous material in the construction of distilling plants is indicated in Fig. 20, (photographed in May, 1938), which shows the condition of the interior of the steel-plate shell of the combined distilling-condenser unit installed on board the destroyer U.S.S. *Clark* (DD361). It should be mentioned that this extensive corrosion took place within less than 2 years after completion date of the vessel and occurred on the fresh-water side of the unit, since the sea water is circulated through the tubes. The condition of the combined distilling-condenser units installed on the other destroyers of this class was essentially the same, indicating that this condition was general. As a result of this experience, the Bureau of Ships authorized the replacement of all steel-plate distilling-condenser shells aboard the DD356-363 class destroyers and aboard the U.S.S. *Brooklyn* (CL40) class cruisers with shells of cast-bronze construction, and in addition required nonferrous construction for all future distilling-plant installations.

Figs. 21, 22, and 23 illustrate the progressive degree of tube fouling of the air-ejector after condenser, inner feed heater and vapor feed heater, respectively, comprising the No. 1 distilling plant installed on board the cruiser U.S.S. *Brooklyn* (CL40). These revealing views were taken in September, 1944, before the installation of the cornstarch-boiler compound evaporator-feed treatment system. The total months in operation since the last cleaning of the after condenser, inner feed heater and vapor feed heater were 20, 24, and 19, respectively, out of which 11.7 months constituted operation in Mediterranean waters.

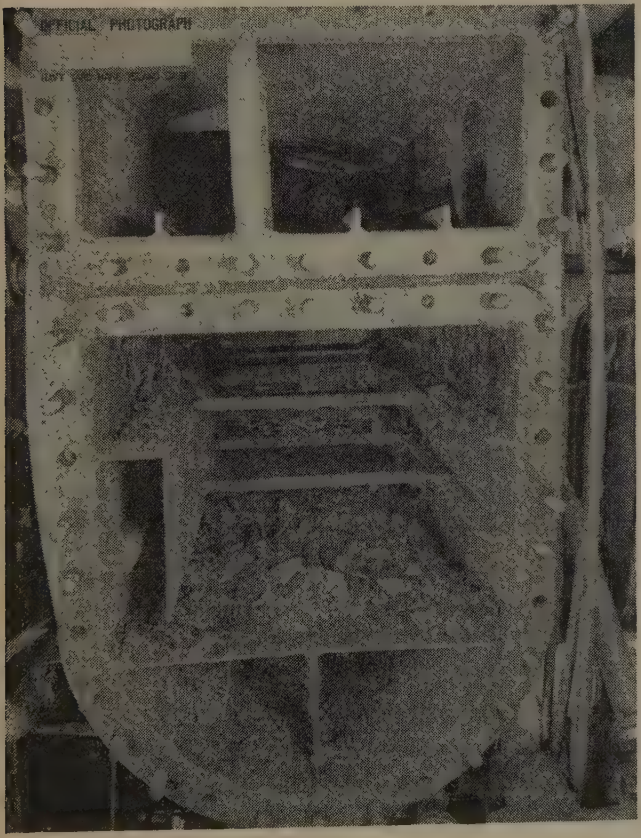


Fig. 20 Condition of Interior of Shell of Distilling Condenser—U.S.S. "Clark" (DD361)

Evaporator Feed-Treatment System

During the summer of 1944, extensive tests to determine the value of pretreating evaporator feed with cornstarch and Navy boiler compound were conducted on distilling plants installed on board the destroyer tenders U.S.S. *Altair* (AD11) and U.S.S. *Hamul* (AD20). These tests were conducted during anchorage at Great Sound, Bermuda, under the direct supervision of Mr. Ray L. Clapper, head of the Service Department of the Griscom-Russell Company. The sea water in Bermuda Harbor was found to be 1.25 thirty-seconds density and supersaturated with calcium carbonate. Before the tests were started, the distilling plants were cleaned thoroughly and placed in a satisfactory operating condition.

The results of these tests indicated that the capacity of the plant operating on straight sea-water feed was about equal to the

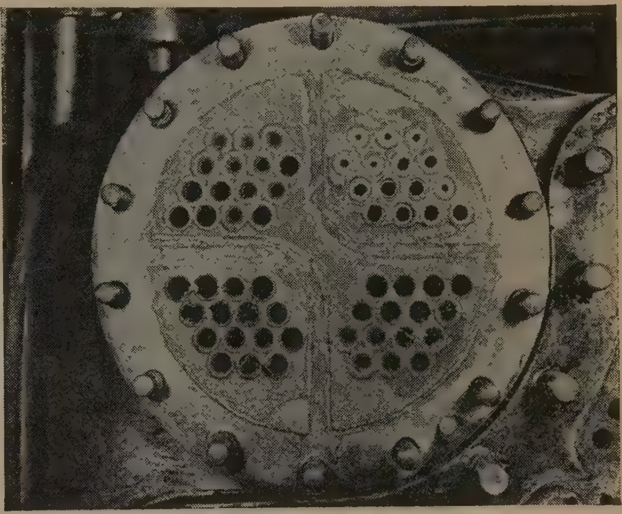


Fig. 21 Tube Fouling of Air-Ejector After Condenser—U.S.S. "Brooklyn" (CL40)

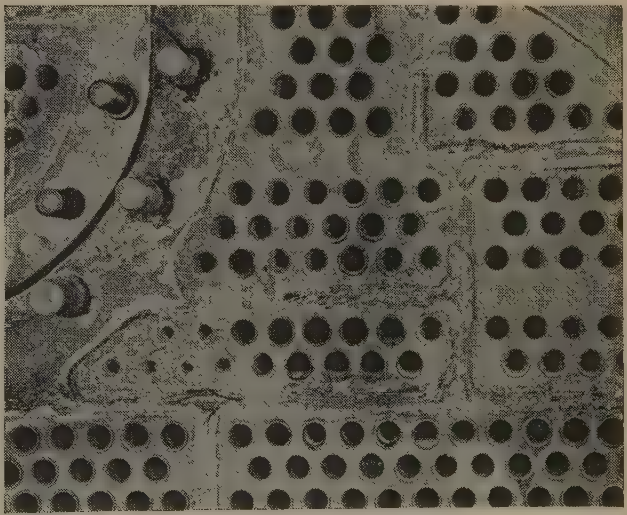


Fig. 22 Tube Fouling of Inner Feed Heater—U.S.S. "Brooklyn" (CL40)

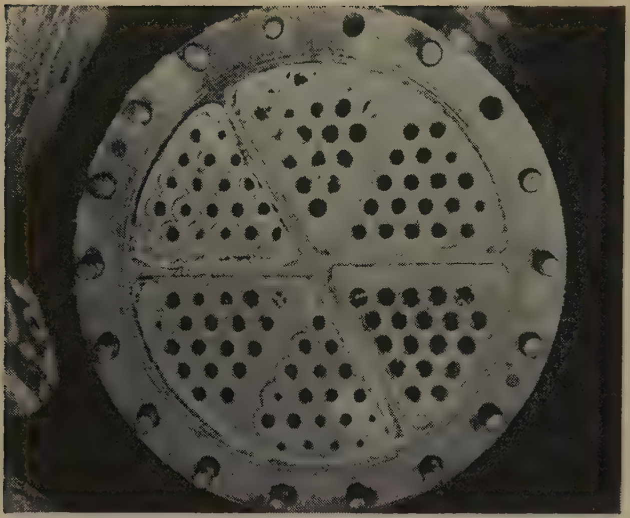


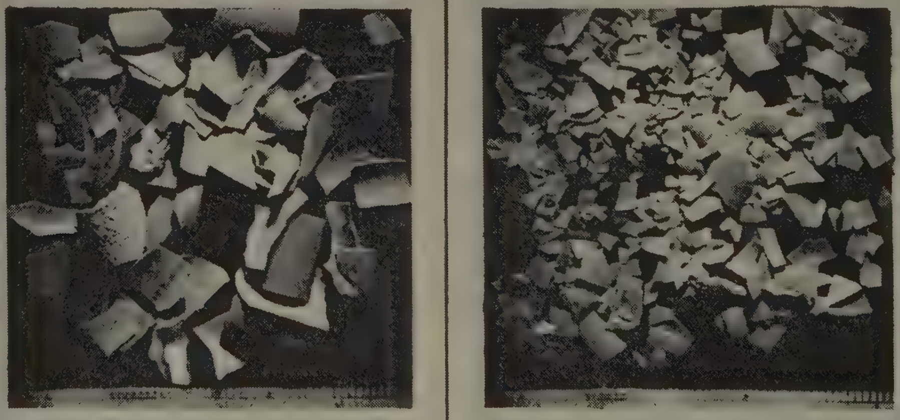
Fig. 23 Tube Fouling of Vapor Feed Heater—U.S.S. "Brooklyn" (CI 40)

plant operating with a pretreatment of 0.2 lb cornstarch plus 0.2 lb boiler compound per 1000 gal up until 465,000 gal of water were distilled, after which the capacity of the plant employing pretreatment of feed exceeded that of the plant operating on straight sea-water feed. However, the rate of heat transfer in gallons per hour divided by the temperature difference for both tests varied between 14.2 at the start and an average of 11.2 at the end of the production of 600,000 gal, equivalent to a 30-day period, thus resulting in a fairly flat rate of heat-transfer versus production characteristic.

Operation with untreated evaporator feed yielded a brittle, porous, and easily cracked scale; whereas, in the case of the plant operated with pretreated feed, there was no scale formation but instead an even coating of white powder was formed on the tube surface, principally on the first effect. The use of all boiler compound (0.4 lb per 1000 gal) resulted in the same conditions just outlined for the cornstarch-boiler compound pretreatment test. The use of all cornstarch (0.4 lb per 1000 gal) was not effective and normal capacity could not be maintained owing to rapid scale formation on all three tube nests. Also, the introduction of the feed-treatment compound through the tubes of the distilling con-

U. S. S. ALTAIR - #1 Evaporator - 1st effect scale from tubes after 340,000 gal. No treatment used.

U. S. S. ALTAIR - #1 Evaporator - 1st eff. scale from tubes after 600,000 gal. No treatment used.



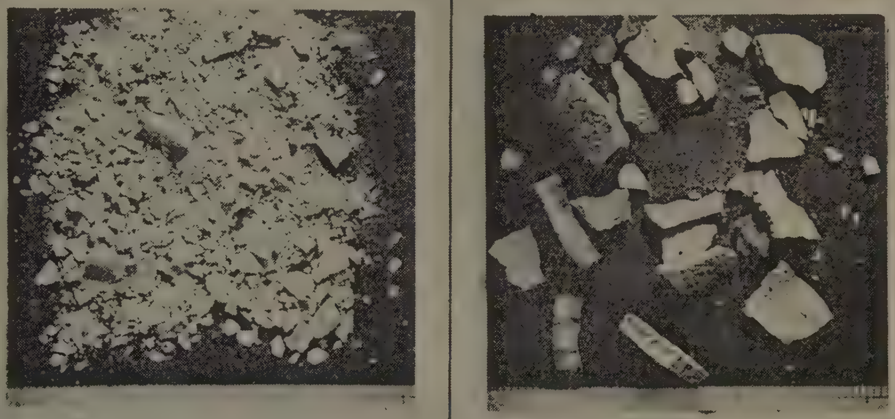
U. S. S. ALTAIR - #1 Evaporator - 2nd effect scale from tubes after 600,000 gal. No treatment used.

U. S. S. ALTAIR - #1 Evaporator 3rd effect scale from tubes after 600,000 gal. No treatment used.

Fig. 24 Samples of Scales Removed From No. 1 Evaporator—U.S.S. "Altair" (AD11)
(Scale divisions at bottom of photograph = 0.1 in.)

U. S. S. HAMUL - #4 Evaporator - 1st effect scale from tubes after 370,000 gal., using .4 lb. cornstarch and .4 lb. Navy boiler compound per 1,000 gal. produced.

U. S. S. HAMUL - #4 Evaporator - 2nd effect scale from tubes after 600,000 gal., using .4 lb. cornstarch and .4 lb. Navy boiler compound per 1,000 gal. produced.



U. S. S. HAMUL - #4 Evaporator - 3rd effect scale from tubes after 600,000 gal., using .4 lb. cornstarch and .4 lb. Navy boiler compound per 1,000 gal. produced.

U. S. S. HAMUL - #3 Evaporator - burned scale from cross-over feed pipes between 1st and 2nd effects and between 2nd and 3rd effects.

FIG. 25 SAMPLES OF SCALE REMOVED FROM NO. 3 AND NO. 4 EVAPORATORS—U.S.S. "HAMUL" (AD20)
(Scale divisions at bottom of photograph = 0.1 in.)

denser and vapor feed heaters was not effective and probably was responsible for the deposition of solids which clogged and heavily coated tube walls with paste material as well as hard scale. Figs. 24, 25, 26, and 27 illustrate the various types of scale and/or deposit accumulated during the afore-mentioned distilling-plant tests.

The Bureau of Ships, in the summer of 1944, authorized the installation of the cornstarch-boiler compound feed-treatment system for low-pressure distilling plants installed aboard all naval vessels in commission and under construction. Reports received from more than fifty vessels indicated the results were beyond all expectations. The mixture was injected into the bottom of the first-effect shell through a perforated pipe by means of "vacuum drag." The introduction of cornstarch-boiler compound not only resulted in the removal of existing scale but retarded scale formation during normal operation of the distilling plant. One destroyer reported that the distilling plant was in operation for nearly 18 months before the unit was shut down for cleaning. An auxiliary vessel reported that existing scale began to fall off after feed treatment was initiated and within a short period better than 80 per cent of the old scale had fallen off, and that the same quantity of water could be distilled in one half the time.

British Experience

Lieut. (E) A. L. Benke, RNR, in a paper before the Institute of Marine Engineers,[2] cited an interesting distilling-plant experience. The distilling plant consisted of two double-effect "soloshell" units installed aboard an American-built escort carrier, having a normal designed output of 44 tons per day.

"In November, 1944, two Admiralty pattern starch-injection units were supplied and fitted, one to each evaporator. Some U. S. Navy boiler compound was added in solution with the starch, in the proportion of 4 lb of starch and 1 lb of compound to each evaporator daily. The result has been that since July, 1944, the tube nests have not required cleaning or scaling. Today there is the finest possible scale only on one effect coil. The ship left home waters on July 7, 1945, and has since opera-

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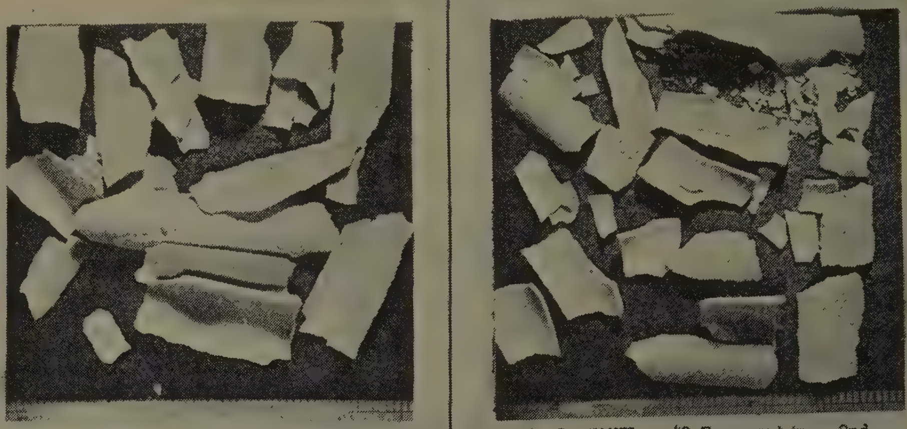
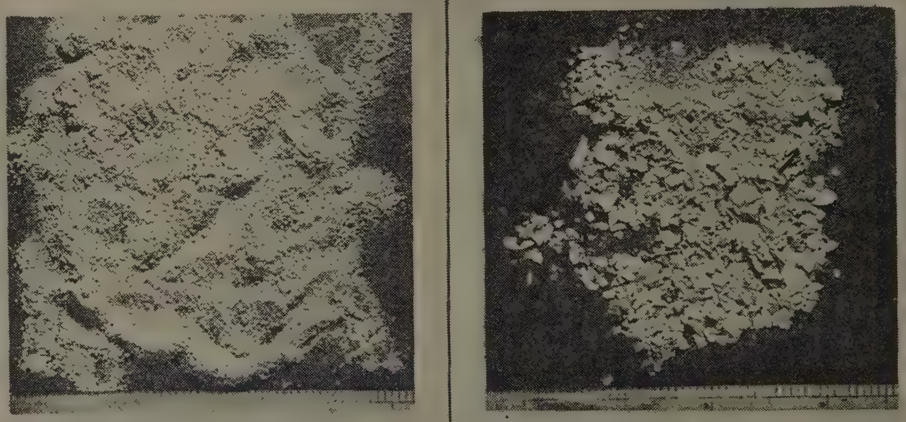
[2] "Starch Injection and Evaporator Output," by A. L. Benke, Proceedings of the Institute of Marine Engineering, vol. lvii, 1946, pp. 161–162.



U. S. S. HAMUL - #3 Evaporator - 1st effect scale from tubes after 143,000 gal., using .4 lb. cornstarch per 1,000 gal. produced.

U. S. S. HAMUL - #3 Evaporator - 2nd effect scale from tubes after 143,000 gal., using .4 lb. cornstarch per 1,000 gal. produced.



U. S. S. HAMUL - #3 Evaporator - 1st effect deposit from tubes after 540,000 gal., using .4 lb. Navy boiler compound per 1,000 gal. produced.

U. S. S. HAMUL - #3 Evaporator - 1st effect light scale removed from tubes after 540,000 gal., using .4 lb. Navy boiler compound per 1,000 gal. produced.

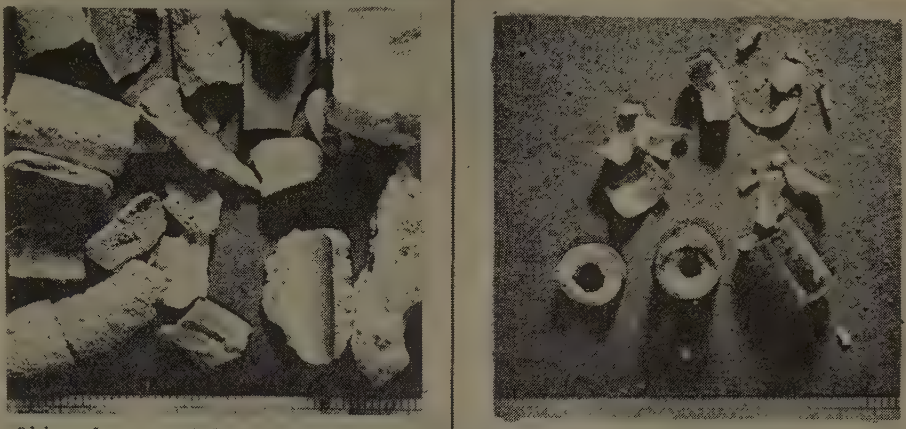
FIG. 26 SAMPLES OF SCALE REMOVED FROM NO. 3 EVAPORATOR—U.S.S. "HAMUL" (AD20)
(Scale divisions at bottom of photograph = 0.1 in.)

ted in waters around the coasts of India, Ceylon, Malaya, Sumatra, and in the China Sea, in all of which waters, temperatures rarely fall below 82 F, and seem to contain more than a normal quantity of scale-forming matter but due to the injection of the starch and boiler compound, none deposits on the tubes. As an indication of the efficiency of the plant operated in this way, a total of 7801 tons have been distilled during an aggregate of 4878 working hours between July 1 and November 10, 1945, which gives an approximate daily average of 77 tons."

The British Admiralty has also experimented with the introduction of acid sodium sulphate and "Belloid T.D.," which is actually the sodium salt of a dinaphyl methane, disulphonic acid, and to date very satisfactory results have been obtained. Full-scale tests were made in standard evaporator equipment aboard the aircraft carrier H.M.S. *Implacable* over the past 2 years. Instead of preventing the deposition of scale, the aim is to control the physical form of the deposited matter so that hard and adherent scale is unlikely. The foregoing tests include comparisons between untreated sea water, sea water plus cornstarch-boiler compound treatment and sea water plus various concentrations of "Belloid T.D." injections The average feed treatment is 12 oz for 10 tons of water distilled. The resultant scale is softer and less adherent, thus permitting it to be easily cracked from the tube surface under conditions of operation of the evaporator.

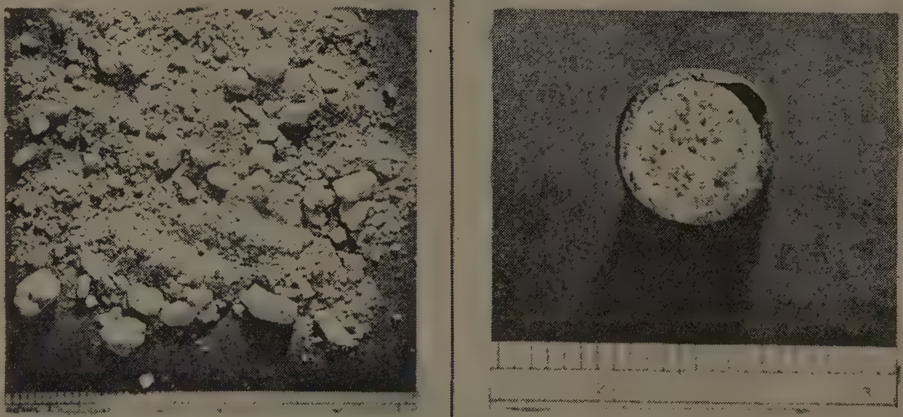
Research Now in Progress

Considerable investigations into the causes and prevention of scale and corrosion in both low-pressure and vapor-compression types of distilling plants, employing sea water as the evaporator feed, are now in progress. The Navy, in collaboration with the Coast Guard, has placed a contract with Armour Research Foundation, whereas the Army has awarded a contract to the University of California for the purpose of trying to find a satisfactory practical solution to this problem. The Bureau of Ships is also investigating other methods of scale prevention, such as a continuous ion-exchange system with a counterflow brine-regenerative arrangement, removal of carbon dioxide, and the installation of a specially designed electrolytic cell in the evaporator-feed circuit.



Old scale removed from evaporator headers and from bottom of tubes - was formed when using cornstarch treatment.

Scale removed from inner surface of 1st effect vapor feed heater tubes - was formed during use of cornstarch treatment.



Scale deposit removed from feed heater tubes in paste form - accumulated during use of cornstarch treatment.

Vapor feed heater tubes solidly plugged with hard scale - formed during use of cornstarch treatment.

FIG. 27 SAMPLES OF SCALE REMOVED FROM AUXILIARY UNITS
(Scale divisions at bottom of photograph = 0.1 in.)

It is anticipated that good results will come out of the various research and development projects concerned with the prevention of evaporator scale, but, in the interim, the Bureau of Ships is requiring the design of vacuum-operated plants, both of the low-pressure and vapor-compression types, for new construction, in order to hold operating temperatures at a minimum.

The Fouling and Cleaning of Surfaces in Unfired Heat Exchangers—Panel Discussion

PREPARED By O. P. BERGELIN, UNIVERSITY OF DELAWARE

This report summarizes the proceedings of the panel discussion meeting of the Symposium on Fouling and Cleaning of Heat Exchange Equipment.[1] Fouling in water circuits, in steam power plants, in marine power plants, in the process industries, and the effect of fouling upon the design of equipment are covered by the papers of the panel leaders. Topics brought up during the general discussion include fouling by calcium-sulphate scale, the suitability of certain "cleanliness factors" used in marine-equipment specifications, the use of additives to reduce fouling by fuel oil, and the effect of manufacturing tolerances and design upon the rate of fouling.

INTRODUCTION

THE fouling of heat-exchange surfaces presents problems of great practical importance, but these problems are so complex that little progress has been made toward their solution. The problems are common to many branches of industry and therefore the Heat Transfer Division of the Society, in its role of furthering progress in the art and science of heat transfer, sponsored a Symposium on Fouling and Cleaning of Heat Exchange Equipment.[1]

The general purpose of the Symposium was to furnish a survey of the field and encourage the presentation of data on fouling. It was felt that such information would be immediately useful to those engaged in the practice of heat transfer, and ultimately would enhance the value of data now being obtained under nonfouling conditions. However, information on fouling is not easy to obtain nor is it ordinarily of a nature to permit generalization or justify publication. In the past it has been accepted generally that surface fouling takes place under such widely varying conditions and involves so many variables that each case should be treated specifically. Most problems have been studied from this point of view, and many techniques have been developed by which fouling can be reduced or the cleaning of fouled surfaces can be facilitated. Further extensive studies will be needed to clarify the many perplexing problems in this field, but it was felt that an important step in the study of fouling would be to bring to light some of the elusive factual data which are known to exist in fragmentary form throughout industry but which have never been published.

The proposal of a Symposium to accomplish this purpose was well received, and a number of papers on fouling were obtained. In addition to the formal papers, a panel discussion was arranged to promote a general interchange of ideas and bring forth additional topics which might not be considered suitable for formal presentation but which would be worthy of discussion. The panel discussion was arranged so that the five panel leaders discussed specific phases of fouling and answered questions pertaining to these phases. During the open discussion the participants recorded their remarks for use in the preparation of a report of the meeting. The present report consists of a summary of the remarks of the panel leaders, a brief coverage of the material presented during the open discussion period, and the written discussion received after the meeting. The presentation is arranged by sections, each covering the topic of a panel leader, including the questions and discussion pertaining to that topic.

Fouling in Water Circuits and Evaporators

By L. F. COLLINS,[2] PANEL LEADER

THE deposits formed from or by "fresh" water constitute the origin of a large number of fouling problems which are similar throughout the process industries. The deposits which originate through precipitation are usually referred to as scales or sludges, while those originated by biological organisms are commonly called slimes or barnacles. More complete definitions are as follows:

(*a*) "Scales" are dense deposits so firmly bonded to their supporting surfaces as to allow virtually no contact between the surrounding fluid and the covered surfaces.

(*b*) "Sludges" are amorphous deposits and seldom are bonded to the supporting surfaces. Thus they are considerably less effective than scales in preventing contact of the surrounding fluid with the surface.

(*c*) "Slimes" are gelatinous deposits resulting from the multiplication of microscopic organisms or the products of their life processes.

(*d*) "Barnacles" are nodulelike deposits having semigranular outer shells bonded to the supporting surfaces and enclosing a slurry of putrifying micro-organisms.

Scales are produced in water circuits through precipitation of mineral solids present in the influent. Sludges may be produced either by the precipitation of minerals present in the influent or by corrosive attack, which usually is originated by chemical reaction between the metal of the confining surfaces and deleterious gases present in the influent. Whether or not scales or sludges will form is dependent upon the nature and quantity of the foreign materials present in the influent, and the physical conditions to which the water is subjected in use. The carbonate and sulphate salts of calcium are the most prolific scale producers because these salts are less soluble in hot water than in cold. In Figs. 1

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[1] Papers presented in this Symposium appear in this issue of the Transactions, pp. 825–869.

[2] Detroit Edison Company, Detroit, Mich.

Panel discussion contributed by the Heat Transfer Division and presented at the Annual Meeting, New York, N. Y., November 28–December 3, 1948, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS. Chairman, H. B. Nottage, Research Associate, American, Society of Heating and Ventilating Engineers, Cleveland, Ohio. Mem. ASME. Recorder, O. P. Bergelin, Associate Professor, Department of Chemical Engineering, University of Delaware, Newark, Del.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society.

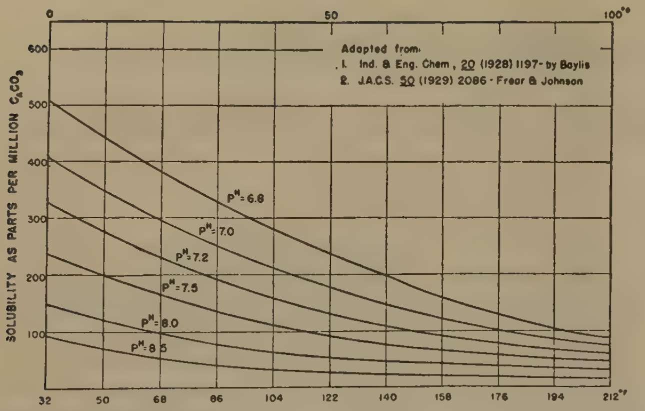

Fig. 1 Solubility of Calcium Carbonate in Distilled Water Containing Carbon Dioxide
(pH values at 73 F, approximately.)

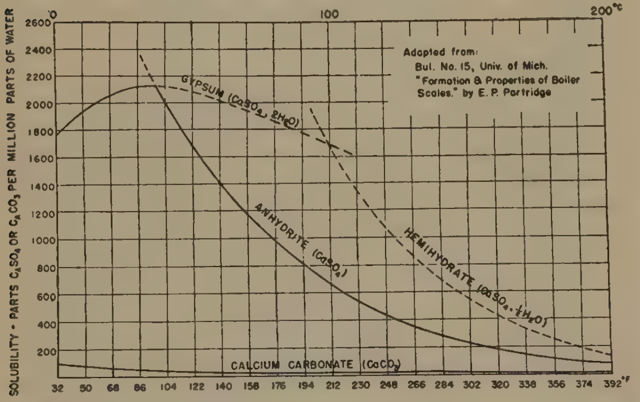

Fig. 2 Solubility of Calcium Sulphate and of Calcium Carbonate for Comparison
($CaCO_3$ in equilibrium with normal CO_2 content of the atmosphere.)

and 2 their solubilities are shown as functions of temperature and pH value.

No comparable criteria have been developed for predicting the sliming potentialities of a given water supply. Most slime-forming organisms are not of the type which cause disease, and thus when a water is subjected to a biochemical analysis for the purpose of evaluating its slime-producing characteristics, tests widely different from sanitary bacteriological tests must be made. Reasonably accurate procedures have been developed for isolating and identifying fouling organisms, but establishing their presence does not establish whether prohibitive amounts of slime will or will not form. As a result it is usually prudent to assume that any natural water is a potential source of slime unless sterilized by chemicals or by heating. A rough classification of the organisms most often responsible for slime is given in Table 1.

TABLE 1 COMMON SLIME FORMERS

Algae

Single celled, sometimes form slimy sheets.
Many celled, form either sheets or fronds.

Fungi

Bacteria (*Schizomycetes*) frequently form slimy surface coatings.
Slime molds (*Myxomycetes*) form slimy sheets as one state of their life history.
Sac fungi (*Ascomycetes*) of which one division, the yeasts, occasionally form slimy aggregates.
Algalike fungi (*Physomycetes*) and stalked fungi (*Basidomycetes*) rarely form slimes but their filaments may hold together the slime of other organisms.

TABLE 2 MINERAL ANALYSES OF MUNICIPAL WATER SUPPLIES (1)

Substance	Unit	[1]	[2]	[3]	[4]	[5]
Silica	SiO_2	2	12	10	9	22
Iron	Fe	0	0	0	0	0
Calcium	Ca	6	36	92	96	3
Magnesium	Mg	1	8	34	27	2
Sodium	Na	2	7	8	183	215
Potassium	K	1	1	1	18	10
Bicarbonate	HCO_3	14	119	339	334	549
Sulphate	SO_4	10	22	84	121	11
Chloride	Cl	2	13	10	280	22
Nitrate	NO_3	1	0	13	0	1
Dissolved solids	...	31	165	434	983	564
Carbonate hardness	$CaCO_3$	12	98	287	274	8
Noncarbonate hardness	$CaSO_4$	5	18	58	54	0

[1] Catskill supply, New York City.
[2] Niagara River (filtered), Niagara Falls, New York.
[3] Well waters, public supply, 30–60 ft, Dayton, Ohio.
[4] Well water, 2090 ft, Maywood, Ill.
[5] Well water, 330 ft, Smithfield, Va.

Note: All values are parts per million of the unit cited to nearest whole number.

Prevention of Fouling in Fresh-Water Circuits

The expedient to be adopted to prevent fouling in a fresh-water circuit depends upon the "type" of water used. The analyses in Table 2 attest the types of water usually available to industry in the United States. The "hardness" of a water is an index of its scaling potentialities, and as shown in Table 2, some waters contain very low concentrations of hardness imparting salts even though they may be highly mineralized (1).[3] Such waters are unlikely to cause scale unless concentrated as by evaporation. For any given water, at temperatures up to about 200 F, the tendency to form a calcium-carbonate scale can be judged by means of the graph and nomogram reproduced in Fig. 3. By means of these data it is also possible to calculate the amount of acid which should be added to a water to thwart the deposition of solid calcium carbonate.

In some instances it is possible to prevent scale deposits by employing "threshold treatments" (2, 3), and dispersive agents. Where temperatures exceed about 200 F, complete softening, as by zeolites or precipitation, provides the only guarantee of freedom from scale. However, complete softening of most natural waters, unless supplemented by air elimination (4), or the use of inhibitors (5), actually may accelerate corrosion to such an extent that fouling will result from the accumulation of corrosion products.

In spray ponds and cooling towers of the open type, especially those having areas accessible to sunlight, light-loving algae growths are likely to cause blocking of the distribution piping and troughs with a slime that is usually stringy in character. For the control of such slimes the chemicals commonly used are chlorine gas, sodium and calcium hypochlorites, the sodium salts of chlorinated phenols, potassium permanganate, and copper sulphate. Neither the phenols nor copper sulphate will remove slime that has already formed, but chlorine gas can be used for this purpose. In open recirculating systems, continuous use of small quantities of chlorine is generally most satisfactory, but in once-through systems where large quantities of water are used, intermittent treatment a few times each day usually will result in satisfactory slime removal and chemical economy. After such cleaning the other chemicals may be used to prevent the re-establishment of slime in the system.

The removal of green algae from a cooling tower should never be used as an indication that the slime-forming organisms on the heat-exchanger surfaces have been removed. The more resistant slime formers which so materially reduce heat-transfer efficiency often will be unaffected by treatment which eliminates algae

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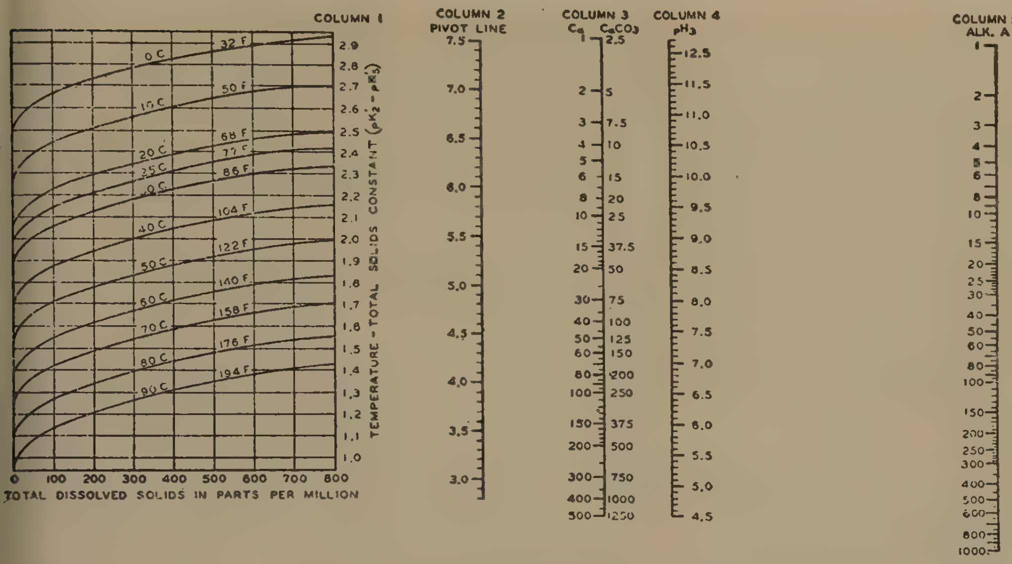
[3] Numbers in parentheses refer to the Bibliography at the end of the Panel Discussions.



(Data required for determining pH saturation.

(1) Total alkalinity, as parts per million of $CaCO_3$.

(2) Calcium in parts per million.

(3) Total dissolved solids, in parts per million.

(4) Temperature, in degrees centigrade, at which pH saturation is desired.

Graph and Nomogram for Determination of pH Saturation by Langelier's Formula

(Applicable within pH Range 7.0—9.5)

*Based on article in Oct. 1936 issue of American Water Works Association Journal and later corrections for Tables 2 and 4. Prepared for Charles P. Hoover of the Columbus, Ohio, Water Softening and Purification Plant by M. L. Riehl.

Instructions for Using Chart:

1. Knowing temperature and total dissolved solids, find temperature and total solids constant on Col. 1.
2. Align this constant with given value of calcium on Col. 3 of Chart, then locate point on Col. 2 of Chart (Pivot Line).
3. Align this point on Pivot Line with given alkalinity on Col. 5; read pH saturation on Col. 4.

Saturation index is pH actual minus pH saturation.

E. G.—pH actual.	pH saturation.	Saturation index.
7.6	8.1	−0.5 (corrosive)
8.4	7.8	+0.6 (scale forming)

Fig. 3 Nomogram for Determination of pH Saturation

completely. In equipment where light is excluded, and this is true of most heat-transfer surfaces, slime formations are due to fungi. Bacteria form a thick soft slime, while yeasts and molds form tough rubbery slimes. Chlorine and hypochlorite solutions fed intermittently are used to prevent such slimes, and only on rare occasions does shock treatment with chlorine fail to yield measurable benefits. While ultraviolet light has been tried as a means to prevent slime formation, no satisfactory installations have been reported.

A considerable amount of research has been done, especially by the marine services, to discover antifouling paints and metals immune to fouling. No information has yet been published indicating that such a paint has been formulated or that a nonfouling metal which is resistant to corrosive attack has been found, although some progress has been made (6), (7).

Discussion

F. M. Alquist.[4] The problem of fouling by solutions containing large amounts of calcium sulphate is still unsolved as shown by the following example: The case in mind was a cone-type steel-shell brine evaporator with 4100 ten-foot-long 2-in. copper tubes rolled into a steel tube sheet. This piece of equipment held 50,000 gal of brine, loaded with calcium sulphate, which deposited a scale that varied from $^1/_2$ to $^3/_4$ in. in thickness. Photographs and inspection showed that there were variations in the thickness and nature of the deposit, but analysis showed it to be 100 per cent $CaSO_4$ with just traces of silica and iron. The deposits were hard and the use of a turbine to cut the scale resulted in cutting the copper tubes. Sometimes as many as 50 to 60 tubes were cut in trying to remove the deposits in this manner.

Various solvents for $CaSO_4$ were studied, but the internal economy of the plant indicated the use of sodium hydroxide which is a fairly good solvent, but not in low concentration as stated in Perry's Handbook. In this case 50 per cent caustic was used and reacted with the scale to form lime and sodium sulphate. A peculiarity of this concentration is that neither the lime nor sodium sulphate is soluble in caustic. In one cleaning test the evaporator was filled with 48 to 50 per cent caustic and kept at 230 F for 5 days with constant circulation. In the first part of the treatment, which lasted 3 days, the concentration of NaOH dropped from 49.6 to 47.6 per cent, and reaction products in the form of suspended particles of lime and Na_2SO_4 amounted to 15 per cent by weight of the total liquid. This suspension was pumped to a settling tank, fresh 50 per cent NaOH solution was put into the evaporator, and the circulation continued for 2 more days. Inspection between stages showed that the deposit had been removed to some extent but the tubes were far from being clean.

The second stage was continued for 2 days, during which the concentration of NaOH dropped to 45.4 per cent. The concentration was then increased to 48.2 per cent by evaporation and more 50 per cent NaOH was added. In the next day and three quarters the concentration dropped to 46.9 per cent. The cleaning was stopped at this point when it was observed that there was a large increase in the amount of sludge in a sample, the amperage to the circulating pump increased from 170 to 220 amp, and there was a temperature difference above and below the tubes indicating poor circulation. The suspension was removed and the increase in solids was found to be 14 per cent, similar to the first stage.

Inspection of the evaporator showed that approximately 90 per cent of the deposits had been removed, but deposits still remained on certain parts of the tubes, indicating that the deposits were not laid down evenly. The same condition exists in boilers where, with the same water, there will be more fouling in one tube than in another. There are of course different temperature conditions, which probably sums up our present knowledge of the problem. The samples of scale taken at the start were 100 per cent $CaSO_4$, but the deposits remaining after the second stage were 85 per cent calcium hydroxide, which is one of the products of the reaction. In the first stage 98,500 lb of solids were removed, and in the second stage 92,000 lb, or a total of 190,500 lb. The evaporation increased 45 per cent to 50 per cent and the rate of steam condensation from 55,000 to 85,000 lbs per hr, indicating that good efficiency was obtained by the cleaning process. However, this method of cleaning is not satisfactory. A better solvent and a shorter cleaning time are desired. The method requires entirely too much outage, and the people who are interested in doing research on fouling still have a problem with calcium sulphate.

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[4] The Dow Chemical Company, Midland, Mich.

E. J. REMSCHEID.[5] Large mercury-arc electric current rectifiers require cooling which is usually furnished by recirculating water through the water jackets of the rectifier and a water-to-water heat exchanger. Corrosion within the recirculating system, which is largely made of iron, is minimized by using distilled water containing 0.1 per cent sodium chromate as an inhibito or by using an approved tap water containing 0.5 per cent sodiu chromate. Ten years of service without appreciable fouling h been reported by installations using distilled water When t water is used, adequate control of fouling is obtained by flushi out the system to remove sludge approximately every 5 years.

The water-to-water exchanger is installed with the raw wat flowing through the tubes in anticipation of considerable fouli from most raw waters. Experience has shown that among t types of deposits which often occur are loose calcium-carbona sludges, hard calcium-sulphate scale, magnesium-dioxide d posits, and voluminous tubercles from corrosion of the iron in t system.

Both mechanical and chemical cleaning have been used su cessfully on the tube side of these heat exchangers. Removab end plates are provided to aid in mechanical cleaning. Chemic cleaning can be accomplished by using either a solution containi 5 per cent by weight of oxalic acid, or a solution of hydrochlor acid with a suitable inhibitor. The solution is circulated throug the tubes at temperatures up to 50 C for rapid cleaning. A traces of acid must be washed out or neutralized after cleanin A solution of $^1/_2$ oz of trisodium phosphate per gallon of water suitable for a wash solution. Care in the original design ar planned maintenance have resulted in successful operation wi raw waters of widely varying cleanliness and mineral content.

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[5] Electrical Engineer, The General Electrical Company, Schenectady, N. Y.

Fouling in Steam Power Plants

By W. L. WEBB,[6] PANEL LEADER

BECAUSE of their effect upon the capacity and economy of steam power plants, plant operators are particularly concerned with the performance of surface condensers, feedwater heaters, and evaporators. Surface condensers usually present the most serious fouling problem, with waterside deposits more common than steamside fouling.

Most feedwater-heater fouling apparently occurs in a feed temperature range of 300 F to 450 F, with most of the fouling on the water side. Fouling of one heater in the cycle results in greater heat absorption in the succeeding heater, thereby increasing the flow from the next higher pressure bleed point at the expense of the lower bleed point. The fouling of the highest pressure feedwater heater in the cycle conceivably might increase the turbine output, but the cycle efficiency always will be affected adversely.

For a given plant, after the economic and practical limits of steam and feedwater quality control have been established, it appears probable that the rates of fouling of both water and steam sides of heaters and the steam side of condensers are beyond the control of the plant operator.

Except in plants where evaporator capacity has little margin over make-up requirements, evaporator fouling on the water side normally presents no serious problem. The crackability of scale produced by evaporator feedwater in some instances has been adjusted satisfactorily by internal chemical treatment. Presoftening the feedwater and employing continuous blowdown usually entirely eliminate scale. The requirement of limiting to a minimum the admission of sludge and scale-forming contaminants to the cycle calls for best possible evaporator vapor quality which, in turn, materially reduces the problem of evaporator-condenser fouling.

NATURE OF DEPOSITS

In condensers the most common waterside deposit is slim usually consisting of cooling-water suspended matter attached the tube walls by algae and bacteria or their gelatinous was products. Periodic sterilization with chlorine or chlorine-co taining compounds usually prevents slime deposits, the contr conditions being dictated by the types of growths encountere In some instances thin hard scales containing silicates of calciu aluminum, and magnesium, iron oxide, and organic matter a encountered. The main constituents of steamside deposits a usually iron oxide and organic matter, frequently includi oil.

Waterside heater deposits are commonly iron oxide, copp oxide, and occasionally oil. The mechanism by which iron oxi is deposited has not been explained fully, but it seems probab that the iron is not in true solution but rather in a colloidal sol tion. Its deposition in certain parts of the cycle seems definite to be influenced by temperature. On the steam side, the d posits usually are similar except for the presence of small pe centages of boiler-water salts. Oil leaving the boiler by stea distillation may undergo partial decomposition and, upon reac ing a heater, contact with iron and copper oxides brings about t formation of heavy metal soaps or metal-organic varnishes. Su coatings deposited on steamside tube surfaces are highly r sistant to conventional cleaning solvents.

In the water phase, iron oxide absorbs oil and deposits it over specific temperature band as a jellylike substance that is co paratively easy to remove. It is perhaps significant that t composition of heater deposits is not influenced greatly by t nature and quantity of the contaminants which tend to produ boiler scale. Ammonia and carbon dioxide can cause seve corrosion, and normally, the proper protection is to get rid of t gases before they enter the feedwater heaters. Ordinarily t can be done, where evaporators are used, by degassing the water thoroughly before it enters the evaporator. A plant deaerator also helps in removing gases. Of course during the operating period, the feedwater heaters must be vented thoroughly to remove these gases from isolated spots back of baffles, etc. Generally, this type of corrosion takes place in dead spots where the gases can accumulate and remain while the reaction takes place.

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[6] Mechanical Engineering Division, American Gas & Electric Service Corporation, New York, N. Y.

METHODS OF CLEANING

The selection of the mechanical or chemical method to be employed for cleaning fouled heat exchangers should be governed by its respective effectiveness, cost, and the equipment outage time necessary. Chemical methods are often the most economical for the waterside of heaters and are substantially the only means of cleaning the steamsides of heaters and condensers. Some types of loosely adherent deposits can be removed effectively from the waterside of condensers by means of metal or bristle tube cleaners, or rubber plugs. In certain instances these may be more economical than chemical cleaning, particularly where the water box and circulating piping require a large volume of solvent in proportion to the area of fouled surface.

Inhibited hydrochloric acid is the customary cleaning solvent, although a pretreatment with a caustic solvent occasionally is required when the percentage of free oil in the deposit is high. Surface-active agents are helpful and, where oil has not progressed to the varnish stage, alkaline oxidizing agents sometimes can be used to advantage before acid-cleaning. Acid-solvent temperatures are adjusted to a maximum of 150 F, and 120 F, where steel and cast-iron surfaces, respectively, are involved. Acid concentrations and contact periods customarily are based upon laboratory solvency tests, preferably upon representative tube specimens with deposits in place. The effect of acid-cleaning upon the ultimate tube life has not been determined fully, but in one particular case eight or more acid cleanings of feedwater heaters apparently have had no effect upon tube life.

In cleaning exchangers chemically, special precautions must be taken against injury to equipment and personnel from contact with the solvents and from explosions of evolved gases. The vents from the water sides of condensers should be carried out of the building, and rapid air circulation provided above the solvent level on the steam side by opening manheads and atmospheric relief valves, and inducing upward air flow in the exhaust stacks by means of a steam jet.

ESTABLISHING THE NEED FOR CLEANING

When new feedwater heaters and condensers are installed, their performance characteristics should be checked while the equipment is known to be relatively clean. Thus a bogey can be set up to which subsequent performance data can be compared. Unfortunately, the urgency of getting new equipment in service usually precludes such tests. Lacking such initial test data, a reliable bogey can be established from specification sheets embodying TEMA standards for heat transfer, or upon the basis of best obtainable performance. The latter is particularly true of old units which, after being thoroughly cleaned, as demonstrated by examination of representative tube specimens, may still give poor heat transfer in terms of current TEMA standards. Stage heaters similarly may operate for long periods at terminal differences somewhat greater than design, even though liberal allowance is made in heat-transfer surface to permit some fouling.

Feedwater heaters are cleaned when the terminal temperature differences become excessive. Although means of by-passing the high-pressure heaters is customarily provided, heater cleaning usually is fitted into the schedule when more important work calls for a major outage. In some of the newer installations, permanent facilities are provided for cleaning the high-pressure heaters chemically.

In order to allow for some fouling of heat-transfer surfaces, without affecting plant performance seriously, most modern condensers are designed on the basis of a heat-transfer rate equivalent to 85 per cent of the rate with commercially clean tubes. Actual performance is often 90 per cent or less of design value.

In routine practice, the condenser vacuum is compared with expected vacuum on the basis of condensate flow, vacuum for 85 per cent tube cleanliness, and rated cooling-water flow. Should the difference between actual and expected vacuums exceed the value specified for the particular unit, test data are taken to determine the heat-transfer rate which then is compared with the accepted standard for the unit. Based on previous knowledge of probable vacuum gains and their duration for specific methods of cleaning, a comparison of the costs of cleaning, and the value of unit outage time with the fuel savings, and the probable increase in capacity, will permit a decision as to whether cleaning is justified and what means can be employed economically.

If a condenser becomes fouled to the extent that the vacuum is reduced by 1 in. Hg, the heat requirement will be increased 200 to 500 Btu per kwhr. If a heater becomes fouled such that the feed temperature to the boiler is decreased 1 deg F, the heat requirement will be increased 1 to 2 Btu per kwhr. These rough rules indicate the effects of fouling.

Fouling in Marine Service

BY D. F. KINERT,[7] PANEL LEADER

AN operating engineer in the Fast Carrier Task Forces has ample opportunity to observe the practical operation of modern marine heat-exchange equipment. The reliable operation of this equipment unquestionably has a direct bearing on the ability of the ship's engineering department to keep the Admiral, the Captain, and the crew in a desirable state of contentment and to maintain the morale of the engineer officer at a suitable level. The distilling plant is without doubt one of the key factors in this problem. There are no sadder words than those announcing water-conservation measures, resulting from the failure of the evaporators to keep up with demand.

In general, heat-transfer equipment in marine practice chiefly handles condensing steam at various pressures, fresh water, salt water, fuel oil, lubricating oil, compressed air, or refrigerants. Of these fluids, fuel oil, and salt water are generally the most troublesome. While the steamsides of exchangers occasionally are contaminated with oil, it is generally susceptible to removal by simple boiling-out processes employing detergents. The corrosion of steamside surfaces is not usually significant. Fresh water on board ship is reasonably free of hard scale-forming salts because of its production from sea water through distillation methods. Where fresh water is used in indirect internal-combustion-engine cooling systems, the addition of inhibiting agents generally reduces jacket corrosion to a negligible problem. Oil contamination is removed from such units in the same manner as outlined in the foregoing for steam sides of other units. Lubricating oil is a generally clean product, and the temperatures encountered in marine practice are not usually productive of excessive fouling.

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[7] Commander, U.S.N., Bureau of Ships, Navy Department, Washington, D. C. The opinions expressed by Commander Kinert are not to be construed as expressing the official views of the Bureau of Ships, The Navy Department, or the Naval Service.

Major Fouling Problems

Fuel oil is a dirty product full of tars and carbonized particles. The fouling problem in fuel-oil heaters increases sharply with viscosity since this controls the temperature to which the oil must be heated. Low velocities and overheating are particularly troublesome. The lighter grades of fuel offer the distinct advantage of requiring a reduced heat input coupled with considerable solvent action. Heaters are generally vastly overdesigned in respect to these lighter fuels since the heaters must be capable of handling more viscous fuels. As a result, the fouling problem is not ordinarily critical, except when the heaviest oils are used. Chemical-solvent methods can be used periodically to eliminate accumulated deposits, the usual cleaning agent being trichloroethylene.

Salt water is the problem child of the marine heat-exchanger media, being capable of producing corrosion deposits, or depositing directly mineral salts of either the carbonate or sulphate type, and of introducing sea life into the heat exchanger. Of all these effects, the scale deposits in distilling plants have been the major concern. This problem is particularly acute in the vapor-compression type of distilling plant because of the small margin in available temperature difference, and the excessive power consumption when any large amounts of scale are present.

It is not intended, however, to minimize the importance of scale deposits in the conventional double or triple-effect evaporator where, under the best of continuous operating conditions, the average output is reduced considerably below the optimum clean-tube output. For example, in order to maintain capacity, it is customary to employ "cold-shock" treatments on a daily basis to crack off the scale thermally. This routine in itself results in an outage time which reduces the average daily operating time by several per cent in order to maintain hourly production rates at a reasonable figure. Furthermore, cold shock is not entirely effective, thereby requiring outages of considerable magnitude for more complete cleaning by laborious mechanical methods.

Continuous Treatment of Raw Evaporator Feedwater

Some success has been achieved through the continuous treatment of the raw evaporator feedwater by injection of boiler compound, considerably fortified by ordinary cornstarch. This method has shown considerable promise where the treatment has been controlled carefully, but there is a definite sensitivity to the concentration used. For example, a relatively minor excess of starch will result in the deposition of a starch layer on the tubes which can be as detrimental to heat transfer as the scale it eliminates. It is quite probable that more effective treatment agents can be devised.

The characteristics of sea growths make them generally vulnerable to rather small increases in temperature above the normal growth environment, and for this reason, many types of marine heat exchangers are not subject to this source of fouling. Certain shore utility stations near the sea have employed mild chlorination methods to reduce sea-growth effects in surface-condenser systems. The adaptability of this method to sliming problems would appear worthy of further investigation.

Recommended Research on Fouling Problems

The foregoing outline covers briefly the usual fouling problems of marine practice, but does not contribute greatly tow raising this subject to a refined technological art. Treati backed by reams of data on clean-surface performance ha been written concerning practically every important type of he exchange equipment and combination of fluids, but there almost no information available for the prediction of fouling ra their effect on heat transfer, and methods to reduce fouling. A result, in the marine field there has been a general tender among designers to provide generous fouling allowance. addition, for certain equipment in which fouling is most noti able, there have been definite attempts to minimize its occ rence. A typical example is the employment of low operati temperatures in distilling plants to avoid the formation of ha sulphate scales.

From the operator's viewpoint, the designers have done splendid job of insuring reliable performance of marine he exchange equipment in spite of any actual fouling encounter However, from a broader viewpoint the modern naval or mercha vessel can ill afford to have any portion of its installed equipme one pound heavier than absolutely necessary for reliable perfor ance. Excess equipment weight imposes a detrimental eff on the available pay load of a vessel whether it is in terms of ra and offensive power, or cargo-carrying capacity. Therefore it recommended that each of the following fields be exploited raise the standards of design knowledge concerning fouling pro lems in heat exchangers:

(*a*) Basic research into the mechanics and chemistry of t various types of fouling, particularly as to the effect of surfa temperature, velocity of flow and heat-transfer rates for the va ous heat-transfer fluids, upon the change of heat-transfer c efficients with operating time. Essentially, this comprises application of a time factor to basic heat-transfer data.

(*b*) Translation of these basic data into a form suitable f practical design work with a particular view of establishing re sonable fouling allowances.

(*c*) Investigation of practical measures to reduce the rate deposition of fouling deposits through the employment of suitab design parameters as regards fluid velocities, heat-transfer rate and equipment configuration.

(*d*) The further development and application of inhibiti agents to heat-transfer fluids.

(*e*) The further development of simple, economical, and quic means of cleaning heat exchangers with maximum safety to t equipment and operating personnel.

Summarized Discussion

A. Devore.[8] How can you reconcile the otherwise general conservative practices in the marine field with the "percenta fouling" usually appearing on United States Coast Guard an United States Maritime Commission specifications? F example, fuel-oil heaters take a 50 per cent cleanliness facto while feedwater heaters generally call for 90 per cent clean ness factor. For conservative practice these figures should b reversed.

Commander Kinert. In my experience I do not think agree with you. Generally, I have noted that fuel oil was muc more of a fouling agent than straight fresh feedwater. O condition to be considered is that the operator has better contr over the feedwater heaters than the oil heaters which must hand whatever oil is available.

A. Devore. In the case of a straight-tube fuel-oil heate given an allowable pressure drop with an order of magnitude of 1 to 20 psi, an oil-film heat-transfer coefficient in the neighborhoo of 20 or 30 is obtainable for the common grade of fuel oil, Bunke

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[8] Chemical Engineer, Davis Engineering Company, Elizabeth, N.

Ordinarily, for industrial work you would apply a fouling ctor or dirt factor of 200, in other words, a correction of 0.005 the magnitude of the over-all resistance. Now, if you take 20 d simply halve it, you are cutting the coefficient down to 10, hereas if you apply a correction factor of 0.005, you might cut it wn to 18 or 19. In the case of a feedwater heater where at mes you have water-film coefficients in the neighborhood of 1000, you have a fouling factor of 2000, you are cutting that film wn by approximately 33 per cent, whereas if you take 90 per nt of 1000, you are taking hardly any correction at all on the edwater heater; so you see the two things are reversed.

These specifications were arrived at, supposedly, in order to be aning over on the conservative side, but they have achieved actly the opposite results. I cannot understand why the Mari-me Commission and the Coast Guard do not adopt the common ractices that have been adopted, for example, by TEMA, which re very conservative.

F. G. Feeley, Jr.[9] A factor is involved in this which Mr. rmstrong has given us a lead on, I think. I agree with Com-hander Kinert very strongly that the steamship operator, or in ur case the power-plant operator, has some real control over his edwater, but he has no control whatever over his fuel oil. This eans of course that somehow or other he can contrive to keep is feedwater heaters clean but he cannot do much to keep his oil eaters clean. Therefore he needs a much greater factor of esign. The other point is of course that fuel-oil heaters are lmost invariably made with steel tubes, while feedwater heaters re normally made of some nonferrous metal, particularly in marine work, 70-30 cupro nickel or thereabouts. I believe Mr. Armstrong intimated quite strongly that you could rely on somewhat lesser fouling factors with the nonferrous tubes than you could with the steel tubes, which means that, as I see it, those factors are somewhere nearly in line with the anticipated operating conditions.

A. Devore. I should like to comment on Mr. Feeley's remarks. His contention is that a much greater factor of design is required to keep a fuel-oil heater operating than would be required for a feedwater heater. True! There is nothing in my remarks implying anything else. TEMA specifies a dirt factor of 0.005 for fuel oil and 0.0005 for boiler feedwater (velocity 3 fps). Thus the magnitude of the dirt factor for oil is 10 times that for the water. On the other hand, use of 50 per cent CF for the oil and 90 per cent for the feedwater accomplishes the following undesirable results:

1 The fuel-oil heater is too large and therefore too expensive. The resulting low velocities cause greater fouling rates so that little, if any, time is gained between shutdowns.

2. The feedwater heater is undersurfaced because the 90 per cent CF does not even come close to decreasing the over-all coefficient to the same degree that a dirt factor of 0.0005 would.

The argument that there is very little, if any, control over the type of fuel oil is irrelevant. The heater already has been designed for the worst possible condition by assuming that the oil handled is the most viscous grade, Bunker C. The argument that ferrous metals rather than nonferrous metals are employed to handle these oils is also irrelevant, as the heavily weighted dirt factor of 0.005 given by TEMA is based on actual field experience.

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[9] Mechanical Engineer, Union Carbide and Carbon Corporation, ew York, N. Y. Mem. ASME.

Fouling in the Process Industries

By R. L. CLAPPER,[10] PANEL LEADER

THE problems caused by the fouling and cleaning of heat exchangers in the process industries are not simple ones and varrant extensive research both in design and operation of equip-nent. The fouling factors proposed by TEMA, based upon field lata and good judgment of various designers, are quite commonly ised and generally will provide reasonable cleaning periods withut abnormal interruption of the process. However, abnormal lirt factors are often encountered when least expected, and are npredictable and cannot be accounted for in design. They may e due to conditions beyond the control of the designer and may esult in the unbalanced operation of a process with a considerale loss of total production or a loss in quality of the product. Iany operators are not accustomed to think in terms of dirt facors and often believe that a unit is reasonably clean when in eality the dirt accumulation may have reduced the over-all ransfer rate to a value of 30 to 50 per cent of that guaranteed. n many such cases the unit is then thought to be too small for he duty.

Causes of Abnormal Dirt Factors

Some of the principal causes of abnormal dirt factors are as ollows:

Fouling Before Exchanger Is Used. Fouling may be due to nproper protection of the heating surfaces before the time when t is placed in operation. Steel parts will corrode rapidly if mois-ure is allowed to accumulate in the unit. Protective coatings may be applied to the surfaces, but care must be taken to remove the coating thoroughly before operation begins. No protective coating is known which will prevent fouling during operation and still give a satisfactory heat-transfer coefficient.

A satisfactory method of protection is to air-dry all surfaces and seal the unit immediately after fabrication. Bags of moisture-absorbing chemicals may be placed inside the unit to take care of any moisture that penetrates the seal. The connections remain sealed until the unit is installed.

Poor Design and Arrangement. Heat exchangers should be specially designed when liquids or vapors are known to contain sediment or to have coking characteristics. Sediment or coke will quickly settle out in low-velocity areas, particularly around baffles and heads of tubular bundles, and also in the tubes themselves when small-diameter tubes are used. This condition may be encountered in processes where liquids normally carry solids in suspension, such as clay for gasoline treatment, wax-bearing oils, char water in paper industries, vegetable oils, and heavy fuel oils.

In designing for such conditions, it is desirable to force liquids through as few parallel streams as possible in order to maintain high and continuous velocities throughout the exchanger. Permissible pressure loss often prevents the best possible design, and it would be well if, in these cases, the user could be prevailed upon to provide higher pumping heads. We have records of properly designed units, operating continuously for more than 1 year on heavy-duty coking-characteristic fluids, where a conventional tubular exchanger would require cleaning within only a few weeks on similar duty. A photograph and a section of one type of unit

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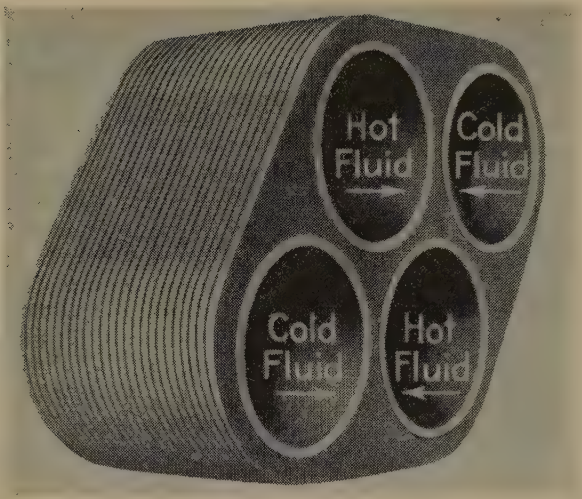
[10] Head of Service Department, The Griscom-Russell Company, ew York, N. Y.



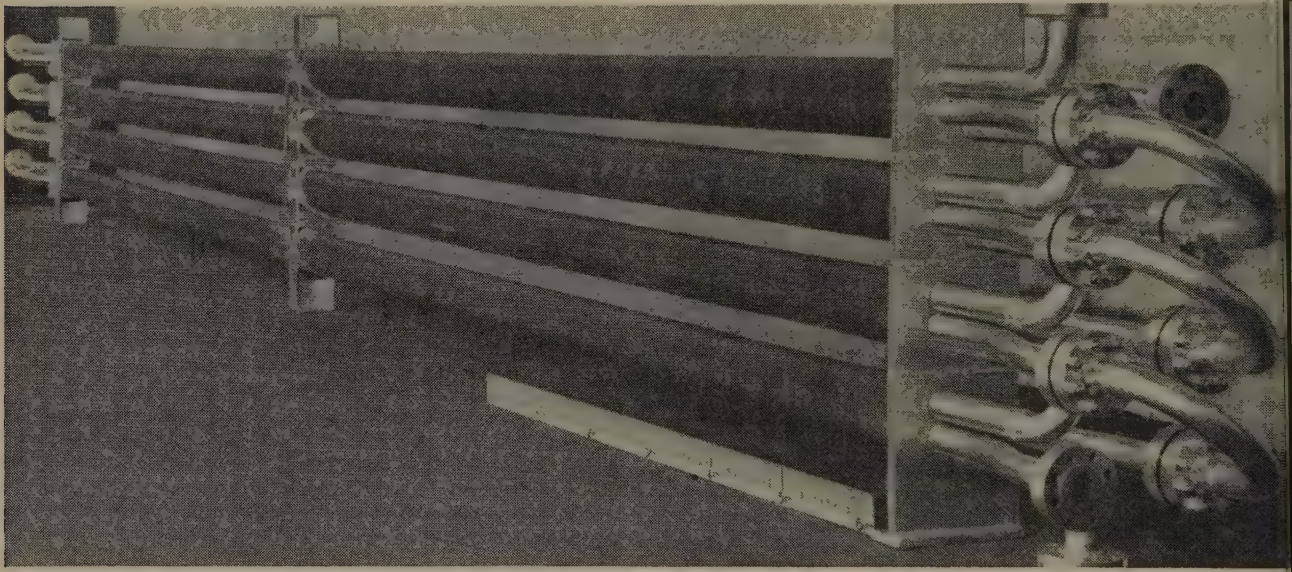


Fig. 4 (*Left above and center*)—One Type of Exchanger Designed for Conditions of Severe Fouling
(Courtesy of The Griscom-Russell Company.)

designed to combat high rates of fouling are shown in Fig. 4.

The installation of units in parallel in order to simplify piping or make use of a single pump often causes rapid fouling in some of the units. In one particular installation, 23 heat exchangers, involving several duties, took water from a common header and discharged into a common header so that there was a fixed pressure drop across all units. Scale accumulated in the hotter units and caused lower water circulation, which in turn accelerated the formation of scale in these units. After several months some tubes in the hotter units were completely plugged while the cooler units were still in good condition.

Decrease in Liquid Flow. Generally, exchangers are designed large enough to meet the maximum operating conditions. As an example, a jacket-water cooler must be able to meet the maximum cooling requirements with the maximum available water temperature. This unit would then be considerably oversized for winter operation, and under these conditions, the operator might decide to throttle the flow of cooling water in order to maintain a reasonable water temperature to the engine jackets. Such operation will result in high tube-wall temperatures with consequent scale deposits and, due to the low water velocities, it also may result in rapid sediment deposits if river w is used. A reduction of the rate of heat transfer to one hal that guaranteed is easily possible. The fouling usually is detected by the operator until the maximum duties of the changers are required during the summer when the maxim cooling-water temperature occurs.

To retard the prevention of scale it is necessary to have eno water to keep the temperature from rising too high. In gene on sea water the outlet temperature from an exchanger sho not be over 110 F, and for normal fresh water not over 120 F. prevent deposits of mud and silt in a water-cooled unit, the ve ity should be above 4 fps second and usually, depending u design conditions and the allowable friction loss, the veloc will be less than 9 fps.

In condensing units, rapid deposition of materials from the pr ess may occur on the outer tube surfaces due to the chilling eff of cold cooling water in winter In oil-cooler units a dewax of the oil may occur owing to low cooling-water temperature, often the wax will remain on the surface even after warmed

culating water is available, so that complete chemical cleaning required to restore the cooling capacity of the unit. These conc tions cannot be corrected fully by exchanger design, but multip small units may be used to advantage since one or more the small units may be by-passed as the cooling-water temperatu decreases. Full circulation is thus maintained through the uni in use and prevents sediment deposits. The most desirab method to allow for changes in cooling-water temperature is recirculation of water in order to maintain the inlet-water temper ture near that for which the unit was designed. Recirculatio however, generally requires additional pumping and contr equipment which complicates the piping system and increases t cost of the installation.

Excess deposits of carbon from heavy fuel oils often acc mulate in oil heaters, particularly in installations where intermi tent heating duties are required. At reasonably high oil veloc ties and continuous flow, the carbon particles will carry along wi the oil, but at low or no velocity and high oil temperatures, th carbon particles accumulate rapidly on the tube surface. Th condition can be avoided by careful regulation of the oil ten perature and by closing the steam supply entirely when the o ow is stopped. A properly designed temperature-control valve nd thermostatic element will accomplish this purpose.

Excess Pretreatment of Process Fluids. Vapors, gases, and quids are often treated before entering heat exchangers to pre- ent corrosion or to remove undesirable materials. In many cases n excess amount of treatment is used, with the result that ex- eedingly heavy dirt or sludge deposits accumulate on the ex- hanger surfaces, decreasing the over-all heat-transfer rate to a reater degree than expected and sometimes causing concen- ated corrosion of the metal. There are cases on record where ithin 5 months a sludge from pretreatment for oxygen elimina- on reduced the over-all heat-transfer rate of boiler feed heaters one third of that guaranteed. In natural-gas treatment simi- r heavy sludges appear in some of the exchangers and have een known to reduce the heat-transfer rate to less than one half hat guaranteed.

CLEANING EXCHANGERS

The removal of the expected normal dirt accumulation is enerally done, either mechanically or chemically, during the egular shutdown periods of the entire process. For exchangers here dirt is expected to accumulate rapidly, multiple units should e installed and by-pass lines provided, so the entire process will ot be interrupted. However, the installation of scrubbers and ilters, and better handling of treating systems will in most cases liminate excessive deposits. Cleaning preparations for the bnormal, unexpected dirts are generally overlooked until the peration of the plant becomes unbalanced, at which time make- hift and expensive cleaning procedures are followed in order to eep the plant in operation. In many cases the exchangers are ot provided with by-pass lines, and an entire process unit must e taken out of service in order to clean a single heat exchanger.

Sediment deposits often can be removed by backwashing with high-velocity water combined with air agitation. With proper preparation of the piping for convenient drainage of liquids from the unit, and arrangements for circulating cleaning solutions and water, the deposition of sediment or solids can be easily kept under control. However, the operators must first be educated to know the effect that dirt accumulation has upon the operation, and how often to clean the unit in order to maintain satisfactory over-all heat-transfer rates.

When the dirt is of such nature that washing and air agitation are ineffective, acid-cleaning is often satisfactory and desirable. However, it must be done at intervals frequent enough to prevent complete stoppage of tubes or shell areas and should be handled by personnel experienced in its use, in order to protect metals and to obtain the best results. When the tubes become completely plugged, acid-cleaning is not effective, and each plugged tube must be drilled in order to remove the deposit.

Table 3 summarizes exchanger duties where heavy dirt accumulations may occur. When exchangers are first installed, sufficient data should be taken to establish the expected temperatures and pressure drops. Similar data recorded at regular intervals by the operator will serve to detect any rapid accumulation of sediment or scale, and from such data the proper interval between cleanings can be established.

Discussion

R. C. ULMER.[11] Mr. Clapper makes considerable reference to some of the deposits that have been found in fuel-oil heaters and other oil heat exchangers. Also, Commander Kinert mentions

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[11] Technical Director, E. F. Drew and Company, Inc., New York, N. Y. Mem. ASME.

TABLE 3 TYPICAL FOULING AND CLEANING PROBLEMS IN PROCESS INDUSTRIES

Process	Duty	Fouling material	Cleaning method	To avoid excess dirt accumulation
utomobile plant	Quench oil cooling	Mill scale	Mechanical	Install filter before cooler. Streamline[a] liquid flow passages.
asoline plant	Vapor condensing Gasoline cooling	Sludge, scale	Air agitate and backwash chemically clean Drill completely plugged tubes	Avoid many units in parallel. Provide pumping head to maintain correct water flow.
as-treating plant	Solution exchanger	Iron-sulphide sludge	Air agitate and backwash	Remove sediment from gas before treating. Keep liquid velocities high.
	Solution cooling	Scale from water	Acid clean	
ndustrial	Oil heating	Carbon deposit	Circulate solvent	Control oil temperature. Keep oil velocity high at all load conditions.
	Oil cooling	Wax from oil	Chemically clean	Recirculate water to maintain design temperature and velocities.
	Jacket-water cooling	Scale and sludge	Air agitate chemically clean	
oke plant	Vapor condensing	Naphthalene	Dissolve by hot water or steam	Remove naphthalene before condenser. Recirculate water to maintain design temperature and velocities.[a]
aper mill	Char water heating	Suspended paper	Mechanical	Streamline[a] the liquid flow passages; keep velocities high.
ower plant	Feedwater heating	Sludge from treatment	Acid	Avoid overtreatment in feedwater stream. Completely eliminate treatment when possible.
ood products	Vegetable-oil heating Vegetable-oil cooling	Vegetable-oil deposits	Acid	Streamline[a] all space to prevent fouling and contamination of product.
xchanger manufacture	Steel baffles and tubes	Iron oxide	Acid	Air dry all steel surfaces and seal all connections before shipment.
etroleum refinery	Heavy-oil exchanger	Coke deposit	Mechanical	Streamline[a] the flow passages. Keep velocities high.
	Vapor condensing	Products of ammonia treatment	Flush with large volume of water	Avoid overtreatment.
	Oil cooling	Clay	Mechanical	Streamline[a] the flow passages. Keep velocities high.
		Wax	Dissolve with heat	Recirculate water to keep velocities high. Provide required water quantity under all conditions.
		Scale and sludge	Air agitate—acid clean	

[a] "Streamline" as used here, signifies flow passages designed to eliminate or reduce to a minimum sudden changes in velocity or direction of flow, and should ot be confused with "laminar" or "viscous" flow.

that there has been a considerable problem, especially with some of the tarry residues, in heat exchangers and oil heaters.

We have done a little work on the chemistry of this subject and believe that such an approach is necessary to obtain success. One of the first problems was to set up a laboratory piece of equipment which would simulate field conditions. Fig. 5 shows the setup finally decided on. Oil is pumped from the bottom of the storage tank through a steel coil immersed in a constant-temperature oil bath. This coil simulates a fuel-oil heater or an oil heat exchanger. The oil passes out of the steel coil, through a section where there is a thermometer by which the oil temperature is measured, back to the top of the storage tank and is recirculated.

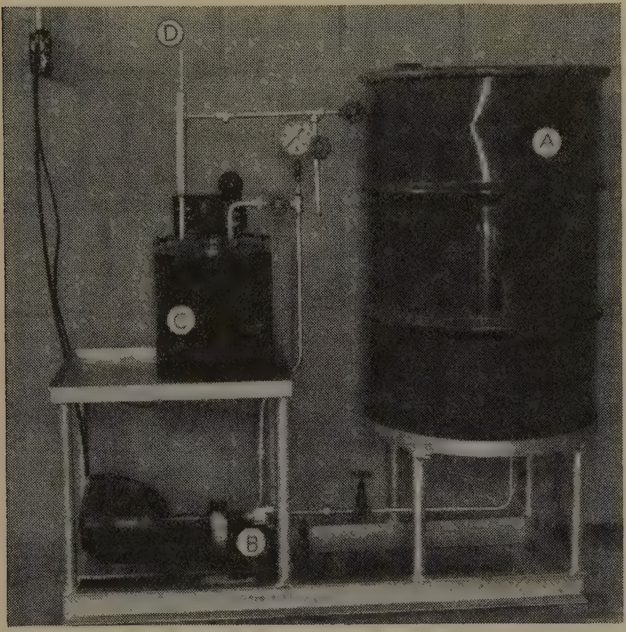

Fig. 5 Equipment Used for Fouling Tests

(A, Storage tank; B, Oil pump; C, Constant-temperature bath; D, Thermometer.)

Oil was circulated through the coil at the rate of 550 ml (0.145 gal) per min, equivalent to a velocity of 26.9 fpm. The volume of oil being circulated was measured accurately, and the rate of pumping maintained constant by throttling the valve in the supply line.

In interpreting results of a run, we reasoned that since oil is circulated through the coil at a constant rate, and since the heat input is constant to the coil, the temperature of the oil leaving the coil should remain constant provided the coil remains clean. On the other hand, if sludge or some other accumulation forms inside of the coil then the temperature of the oil would be expected to drop. Preliminary tests indicated that with proper agitation of the oil in the constant-temperature bath, and by using a good grade of oil, no deposit formed on the outside of the coil. Also, nonferrous metals appeared to act about the same as steel with respect to adherence of sludge.

Fig. 6 shows a typical plot of the temperature of the oil leaving the coil against days. It will be noted that it takes about 8 or 10 days to get the system up to constant temperature, namely, about 165 F. Then for about another 20 days it will be noted that the temperature gradually drops and at the end of about 25 days the drop has leveled off and the temperature is more or less constant. The drop of temperature indicates that deposition of material was taking place inside the coil and decreasing the heat transfer.

Next, a study of the type of deposit that formed in the coil was made. It will be noted that in the papers of this Symposium sludge has been referred to mostly as a tarry residue. We w very much surprised to find that "tar" is not the chief constitu commonly encountered. Rather, the chief material which posits is an oil-water emulsion. Emulsions of this kind may as high as 50 parts of No. 6 or Bunker C oil to 50 parts of wa That such sludges readily form can be demonstrated by a sim laboratory experiment. When oil and water in about the r referred to are stirred very vigorously, the material will set and will not even pour. This is the type of sludge that was fo to be most common from oils in present usage.

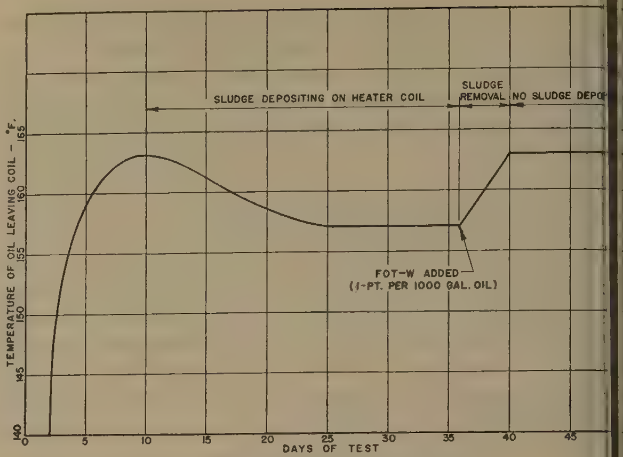

Fig. 6 Fuel-Oil Fouling Test

Much of the emphasis in this Symposium has been on mean removing such deposits from heat exchangers. It would highly desirable to prevent such deposits, and we have given so consideration to preventive measures. We refer to the use of ad tives in the oil during operation, to prevent the deposition of th sludges. In recent years there has been a lot of talk about th additives and some very extravagant claims have been made. the first place, normally only about 1 pint of the additive is ad per thousand gallons. It is a little difficult at first to see h such a small amount of the additive could have much effect, if rather unstable emulsions are present it is a little bit easie understand the action of the additive. Actually, it is not quired to dissolve the deposit but only to break the emulsion.

We made some tests with oil additives and with an effect material obtained a condition such as shown in the right-hand p tion of the curve. When the temperature had "leveled o after some 30 days, an oil additive was added at the rate of 1 p per thousand gallons of oil. It will be noted that in the course several days the oil temperature increased to its original va indicating that the deposit was removed. Note that it was removed immediately; it took a matter of days. We expl this on the basis that the additive actually was breaking emulsion which had deposited on the coil. It will be noted t once removed the additive prevented redeposition of sludge indicated by no decrease in temperature. This indicates th the surface remained clean.

Really, when one considers this situation a little bit further, dosage of 1 pint per thousand gallons of oil is not so unusu For example, it amounts to about 150 ppm. Certainly, for ample, in water treatment, we normally maintain much lo tolerances in the case of phosphate and sulphite and yet we obtain quite marked effectiveness.

There is another way in which the oil additives can be used. refer to a system that is fouled with an oil sludge such as me tioned here and where it may be desirable to clean the equ ment without taking it out of service. In many cases, if the d ge of additive is increased to possibly 10 times the one given, and added ahead of the oil heater or oil heat exchanger, in a matter of possibly a week or even a shorter period, most of the deposit can be removed and the operating characteristics of the heater practically restored to normal conditions. If the heater is out of service, the oil can be drained out, the heater filled with the additive and after 8 to 24 hr, the deposit will be loosened and no further cleaning required. By such methods time can often be saved, and expensive chemical cleaning operations eliminated.

A word of warning perhaps is warranted regarding the use of additives. We have found many to contain high amounts of diluents such as kerosene and fuel oil. Also, most of those which we have encountered will not appreciably attack oil-water emulsions. Specially formulated additives containing specific wetting agents and detergents were found to give the best results.

Influence of Fouling on Design

By R. M. ARMSTRONG,[12] PANEL LEADER

FOULING exerts a major influence on the design of heat exchangers. The five main factors affected are the heat-transfer rate to be used in design, the metal to be used, the size and type of tube to be used, the tube-bundle design, i.e., fixed or floating, and the ease of disassembly for cleaning. Each of these factors will be considered separately.

Heat-Transfer Rate Used in Design

The performance of almost any heat exchanger varies widely over the life of the equipment, but the designer actually can calculate only the clean heat-transfer rate and must estimate the service heat-transfer rate. The design as finally chosen is the designer's best judgment of the amount of fouling to be prepared for in service, with any further fouling to be taken care of by cleaning the equipment. For example, a given liquid heater may operate at an over-all coefficient of 200 when new and about 100 when fully fouled. The designer may decide that 125 is a good value for a service heat-transfer coefficient, on the premise that a value as low as 100 represents bad practice.

Many exchangers, built of stainless or nonfouling metals, operate substantially clean at all times. This may be due to the operation itself being clean in nature, such as cooling clean organic solvents by means of clean water, or it may be due to cleaning the equipment thoroughly between runs, as is done for milk or beer coolers. On the other hand, some exchangers foul substantially very quickly, and thereafter operate regularly in a highly fouled condition. Tests at the University of Illinois on vertical ammonia condensers (8) showed that the performance drops about one third in the first few hours of operation then levels off to a rather stable condition.

For design, the clean rate usually is calculated first and then fouling resistances are applied. The most widely used fouling factors are those of TEMA but most designers also have tables of fouling factors which represent their own experience. In some cases experience, or rule of thumb, has developed combination film rates and fouling factors. For instance, on ammonia or freon condensers the lump sum of condensing film and surface fouling resistance is known, while the breakdown between the two is open to argument. So, in design of these condensers the metal resistance, and waterside film and fouling rates are calculated in the conventional way, but the refrigerant-side film and fouling rates are combined.

The same method is used in the design of freon and ammonia coolers for brine and water where the boiling film rate is combined with the fouling rate. Experience shows the combined value to be correct but the breakdown between the two is not clear. This also may be said for the design of ordinary steam water heaters, where submerged or partial boiling may take place and completely upset conventional calculations.

Metal to Be Used

The TEMA fouling factors do not differentiate between steel, which fouls easily in most cases, and copper or other nonfouling materials. There is, however, a considerable difference in the amount of scaling under the same fluid conditions, when the tube metal is changed from steel to copper or some other nonfouling metal. A gain of as much as 30 per cent may be obtained in a number of heat-transfer operations by merely changing the tube metal from one which supports fouling to one which does not. Normally, in water service at least, steel supports fouling, while stainless steel, copper, brass, monel, cupronickel, and some other metals do not lend themselves easily to the attachment of scale and give much better results. Bimetal tubes can cut down fouling materially where the process fluid demands steel because of corrosion. Their use may allow a higher service heat-transfer rate, resulting in a smaller heat exchanger. In picking metals, the designer also must bear in mind that if the equipment is to be cleaned chemically it must have some resistance to the inhibited acid, caustic solvent, etc., that will be used in the cleaning operation.

During the panel discussion it was suggested that preferential fouling might be induced in a manner similar to preferential corrosion. While such a procedure might be possible it does not appear attractive. For one thing, the amount of material available for fouling usually is so large that a reasonably sized surface would not remove it effectively. Filters or pretreatment seem to be the indicated solution. In one sense the heat-transfer surface itself is an area of preferential fouling due to its difference in temperature, and any other preferential-fouling area would have strong competition for the fouling material.

Size and Type of Tube to Be Used

When equipment operates clean at all times, small tubes and helical coils offer very high heat-transfer rates and compact equipment. If fouling is likely to be encountered, such designs become inadvisable and larger tubes of straight design open at the ends for access are more desirable. A small tube closes up more rapidly with the same amount of fouling, and may lead to very much greater pumping and friction losses compared to design conditions. The helical-coil-type exchanger can be made very efficient and compact for clean applications or for duties where chemical cleaning can be carried out successfully. Where fouling is appreciable the problem is difficult to solve, and inspection of the inside of the tubes is very difficult.

Most finned tubes work best in clean fluids. When fouled, the space between fins becomes filled and the amount of surface in effective contact with the fluid is a much lower percentage than on a bare tube. Finned tubes which have wide spaces between the fins are less affected by fouling, but they are also mostly adapted to low-efficiency operations. U-tube bundles are inexpensive and allow freedom from thermal expansion stresses, but cleaning the inside of the U-bends by mechanical means is very difficult.

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The least difficulty from fouling is to be expected when tubes are straight, smooth surfaced, not over 12 ft in length, at least 1 in. inside diameter, and open for access at each end.

Tube-Bundle Design

In theory, a removable bundle is required for mechanical cleaning of the outside of the tubes, and preferably the tubes should be arranged on a square pitch. In practice, many users cannot remove a tube bundle in the field without a great deal of effort and expense. Bundles are often very heavy, the baffle and shell may be corroded and irregular, and the accumulation of dirt and scale inside the shell may make it very hard to pull the bundle in the field without damage. Therefore in many cases tube bundles are seldom removed and many removable bundle units are built with staggered tube pitch to get more surface in a given shell, even though cleaning is to be done by mechanical means. In some cases it may even be more economical to use a fixed-tube-sheet exchanger and plan to renew the whole exchanger when it is worn out. When a shell is worn the dimensions are no longer the same as when the shell was new. Since it usually is impracticable to measure these dimensions, the replacement tube bundle often does not fit nearly as well in the old shell as the original bundle fitted into the shell when new. In many cases then, the floating bundle can be considered more a feature to take care of thermal expansion stresses than to provide for cleanability of the outside of the tubes.

The flow pattern of the shell-side fluid may sometimes change considerably due to fouling. For instance, the fluid may by-pass through the crack between tube and baffle hole when the unit is clean, thus operating partly in crossflow and partly in "orifice" flow, but when the equipment becomes fouled, these cracks may be stopped up, causing pure crossflow. Few experimental data are available to demonstrate the effect of this change upon the heat-transfer rate, but it is obvious that the change will materially affect the pressure loss in the equipment. When the fouling is of a type that this change is suspected, proper allowance must be made in calculating design pressure loss.

When designing the shell side of an exchanger, some allowance for fouling can be made by placing the pitch of the tube arrangement in a horizontal manner and making the baffle cuts vertical to allow some movement of the fouling material through the shell. Square tube pitch is used and any wide lanes in the tube arrangement are to be avoided.

Ease of Disassembly for Cleaning

Where fouling is immediate and severe, it may be good design to put the fouling medium inside the tubes, design a bundle not over 12 ft long and use quick-opening bonnets. For instance, in heating milk, tomato juice, and other food products, there is appreciable fouling from the start. Such exchangers should have swing bolted or other quick-closure designs allowing quick access for cleaning out the stone or scale every few hours. In other operations, often with pharmaceuticals such as penicillin and streptomycin, the residue left from one batch will ferment and contaminate the next batch if not cleaned out of the tubes between batches. Here also, quick-opening heads are used, and it is desirable to use fairly large tubes for easier cleaning.

Although disassembly is not ordinarily required for chemical cleaning, there remains the problem of inspection after cleaning. The suggestion made during the discussion that easily removable inspection plates be added on the shell and heads is a possible solution, but it would involve considerable expense and each case would have to be considered on this basis.

Cost of Fouling to the User

It is not hard to calculate the saving in operating cost betwee clean exchanger and a dirty one. In practice, however, often only reason a condenser or exchanger is cleaned is that the syst begins to lose its function or capacity so badly that the wh plant is disturbed. Therefore many exchangers are operati as a regular thing at much below their proper rates, and usua the saving to be obtained from cleaning is a good deal more th the cost of periodic cleaning. The heat-exchanger manufactu can do his bit to remedy this situation by making it possible the user to clean the equipment with reasonable ease, and pleading with the user for his own sake to do the cleaning in or to keep efficiency high.

Discussion

A. Wurster.[13] The information presented regarding the fo ing of heat exchangers is of interest and no doubt will stimula future investigations of this subject, but to be of value, such vestigations must be accurate and the data must be converti to other exchangers of the same or similar design.

Fouling factors as now used, while allowing for the type of fo ing, do not give consideration to the tube material. They a do not include any allowance for the individual designs and th effect upon the fouling tendencies. For example, the fouling fa tors given by TEMA are planned to be applicable for gene purposes and for this reason, in many cases, will either be t conservative or will not provide adequately for severe fouli conditions. The point is, that particularly where the fouling on the shell side, it will depend to a great extent upon the effect the design of the exchanger upon the distribution of the fluids the tube surfaces.

In recent years, users in considering fouling problems have come more conscious of the importance of the design of exchange on the shell side. For example, if a shell is bored and the circu ference of each baffle is machined to provide a close fitting tu bundle of known tolerances, fouling can be greatly reduced. T type of design reduces to a minimum the by-passing of the sh fluid between the shell and the circumference of the baffles due large pressure differences caused by the accumulation of slud and dirt on the outside of the tubes. Further, proper baf stripping in the annular space between the tube circle and t inside diameter of the shell will force the fluid into the tube ban and in combination with a close fitting tube bundle, will utili the pressure differences between the baffles to force the fouli materials through the exchanger. This is not the case for exchanger in which the shell is not bored and the tube bundle f loosely and is poorly designed.

On the tube side of an exchanger, fouling is greatly affected the head design. For example, in an exchanger handling a vi cous tar and having a small straight inlet connection, the t will pass freely through the center tubes opposite the connectio while the tubes in the circumferential area will become clogged a short period of time. In this case a side inlet connection w provide much better distribution of the tar into the tubes, with t fouling taking place uniformly and being greatly reduced.

As is generally realized, the velocity of the fluid, whether in t tube or shell side, has a great effect upon its fouling tendenci In this respect large amounts of excess surface may have a detr mental, rather than a beneficial effect. Here the question aris as to the best way to apply this surface; whether to increase t number of tubes and thereby reduce the velocity—or lengthen t tubes, which makes cleaning more difficult. If the coolin medium quantity is controlled by the outlet temperature of t luid being cooled, the excess surface introduced by the fouling actor tends to accelerate the fouling action by reducing the velocity of the cooling medium.

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[13] The Andale Company, Philadelphia, Pa.

Considering the foregoing points, it is obvious that any data btained from tests made for the purpose of determining fouling actors, must include accurate information on the design of the heat exchanger. If the fouling is on the shell side, all design details of the tube bundle must be known, including the tube arrangement, the clearance between the inside diameter of the shell and the outside diameter of the baffles, the type and design of the baffle stripping, the baffle arrangement, the velocity at various points, etc. If the fluid passes through the tubes, the head design and connection arrangement, as well as the fluid velocity, are important. Only by knowing the operating conditions and the type of fouling, as well as the exact exchanger design details, is the information of value and convertible for use in the determination of fouling factors for other equipment.

CLOSURE

The interest displayed throughout the Symposium was gratifying to the sponsors, and it is felt that a worth-while contribution was made in summarizing the present state of knowledge in the field. On the other hand, very little was accomplished in compiling a systematic record of test data on fouling, or on formulating plans for future studies. It is this latter phase of the problem which warrants serious consideration. Progress in the field probably will be slow and will be extremely expensive if tests are to be carried out on large-scale equipment over extended periods. One approach to the problem would be to encourage the compilation of plant data on new exchangers as proposed by several of the panel speakers. These records, however, would not be of general benefit unless published or submitted to some central group for correlation and comparison.

Another approach to the problem is through chemical investigations such as reported by Dr. Ulmer in his discussion. This type of research would be less expensive to maintain but the extent of the field again indicates that the work should be done at several locations and could be handled more efficiently if there were some central advisory group to co-ordinate and encourage the work. Some stimulation appears necessary if, in a reasonable time, the solution of the problem is to advance from the era of fragmentary knowledge, improvised remedies, and unpredictable performance, to a state where principles of design and operation are well established and are able to eliminate or substantially reduce the fouling of heat exchangers.

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